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SALVAGE AND RECOVERY DATA BOOK -- STATIC LIFT FORCES.(U)
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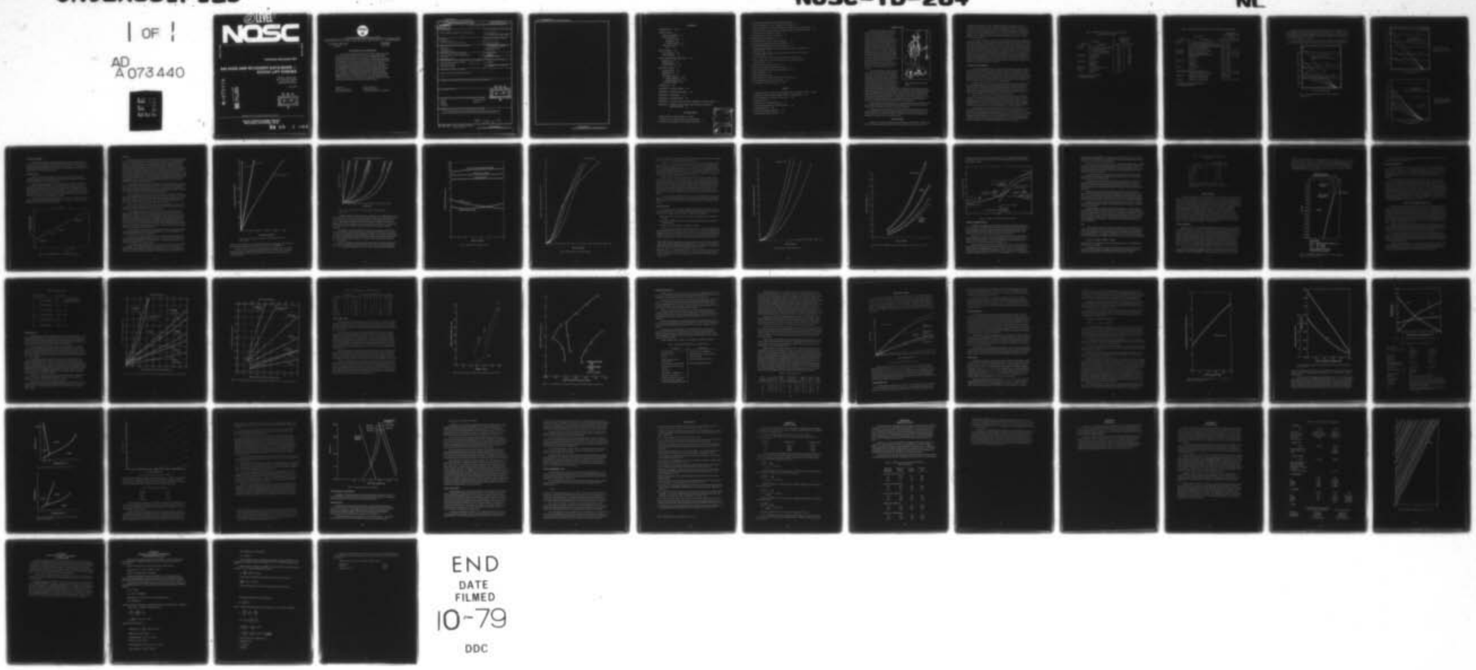

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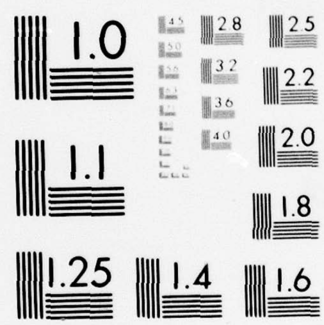

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SALVAGE AND RECOVERY DATA BOOK — STATIC LIFT FORCES

J Muller, Seaco, Inc.
RT Hoffman, NOSC
(Contract Monitor)

June 1979

Approved for public release; distribution unlimited

NAVAL OCEAN SYSTEMS CENTER
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AN ACTIVITY OF THE NAVAL MATERIAL COMMAND

SL GUILLE, CAPT, USN
Commander

HL BLOOD
Technical Director

ADMINISTRATIVE INFORMATION

The Extended Salvage Depth Capability (ESDC) program was begun at the Civil Engineering Laboratory in FY 74 to develop equipment for salvage and recovery of objects to ocean depths of 20,000 feet. Initial development work was done on a self-contained, buoyant lift module which was demonstrated through scale model testing. Later work involved analysis of a wide variety of salvage system components for a broad range of recovery object sizes. During this latter effort, data were accumulated on the methods available for raising objects from great depths. This report is an assemblage of inputs from K. W. Tate of the Civil Engineering Laboratory, D. W. Caudy of Battelle Memorial Institute, Columbus, Ohio, and R. T. Hoffman of the Naval Ocean Systems Center, Hawaii Laboratory. It was authored by Justus Muller of SEACO, Inc., under Contract Number N00123-76-C-0646. It is intended to present, in a coherent manner, data of interest to the deep salvor regarding the capabilities of lift lines, buoyancy gas sources and pontoons, and fixed and variable buoyancy.

Released by
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Ocean Systems Division

Under Authority of
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Environmental Sciences Department

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This data book presents a survey of existing technology available for use in salvage/recovery operations. Data are presented on lift lines, drill strings, buoyant containers and gas sources.

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INTRODUCTION

Figure 1. Nineteenth century diving apparatus.

This handbook is intended to make essential information available to those planning and conducting salvage and recovery operations. Its first objective is to survey current technology. In the past, salvage and recovery were accomplished using rudimentary equipment that was severely limited by the crude technology of the age. Figure 1 is an example. Today, more comprehensive undertakings are conceivable because of the rapid advance of science in a variety of disciplines. Metallurgy has given us steel pipe with a safe working tensile strength in excess of 100,000 pounds per square inch. Chemistry has given us synthetic polymers with a density approaching that of water and a tensile strength approaching that of steel cable. Electronics has given us underwater television. Each has had its effect on salvage and recovery procedures.

The second objective of this handbook is to review and outline safe working practices. In the past, a lack of knowledge concerning safe practice, rather than a lack of appropriate equipment, has sometimes limited what could be accomplished under water. The modern shallow-water diving helmet had been perfected by John and Charles Deane in 1828, and the closed, hardhat diving suit was developed to its modern form by Augustus Siebe in 1840. But the "rheumatism" that crippled so many nineteenth century divers was not brought under satisfactory control until the diving tables of the elder Haldane were introduced in 1906. In our own time, the use of synthetic line for deep lifts from a heaving vessel has solved one problem and created another, less well understood, which will be discussed later. Though what constitutes safe practice is a constantly evolving field of understanding, an attempt has been made in this handbook to construct a conceptual framework in which progress can be discussed.

The final objective of this handbook is to survey possible future extensions of existing technology. Ideas under development in the laboratory, or on which work could be resumed, will be discussed in the context of current capabilities.

This document deals with direct lift by means of steel cable, synthetic cable, and a string of drilling pipe. It addresses pontoons and lift bags, an old salvage technique now more attractive for new materials. Finally, it discusses buoyancy gases, including several gas generation concepts not yet reduced to practice.

LIFTING LINES

Shallow-water salvage operations are often guided by common sense, experience and tradition. The approximate weight of the object is generally known. Breakout force is

estimated, often intuitively. Surface dynamics problems often cannot be forecast accurately. For a quick recovery, necessity often dictates selection of wire ropes from available stock and the most suitable winch or cable-puller at the scene. At times, the range of tide is used to supply the lifting force, perhaps with ballasting and deballasting assistance. Such operations sometimes succeed. In the end, however, more scientific methods tempered with sophisticated understanding usually must be used to conclude the mission successfully.

Conversely, a deep-ocean salvage or recovery operation is almost invariably a race against the elements. This factor, together with the high logistical costs and complications of getting on station and remaining there, demands a highly-engineered approach. If wire rope is used, the actual safety factor must be known. In a very deep lift, the self-weight of the wire rope may make necessary the use of a stepped or tapered line with the thickest line at the top having the necessary strength to support the weight of both the suspended line and the object. Or, a synthetic line may be preferable because of its nearly neutral buoyancy. However, the greater elasticity of synthetic lines can cause other problems. Under certain conditions, the vertical motions of the surface platform are amplified in the lifting line until its breaking stress is exceeded. Dynamic loads and snap loads must be understood, calculated and monitored. In some instances, as will be seen, recovery of the object in two or more stages may be desirable.

STATIC LOAD CAPABILITY

A review of manufacturers' data on the weight and ultimate strength of various types of rope, in air and in water, shows that these properties are approximate constants of the line size. That is, the weight per unit length, either in air or in water, is equal to some constant times the square of the diameter. The same is true for the ultimate tensile strength. Table 1 presents the approximate weight constants for steel wire, nylon, Kevlar, polypropylene, and polyester rope. Table 2 lists the approximate strength constants for these same types of line, and also for dacron, manila and coir.

Normal safe practice on land is to use a safety factor of five, applied to the maximum anticipated static load. The maximum static weight is estimated, including the weight of the line itself, and a line with a breaking strength of at least five times this value is selected. Experience has shown that a safety factor of five is sufficient to keep within the normal working strength of the line while allowing for dynamic loads and reductions in strength at splices, attachments, and points of wear.

Tables 1 and 2 give values of 1.05 pounds per foot-inch-squared, and 65,000 pounds per inch-squared for galvanized 6 X 24 fiber core steel wire in water. Reducing the latter by a factor of five and then dividing by the former gives 12,380 feet as the longest length that should be expected to bear even its own weight in water. A similar calculation for 6 X 37 steel hawser produces a result of 12,167 feet. Thus, 10,000 feet is about the useful limit for steel rope if ordinary land-safe practice is followed.

An alternative to this standard safe practice is to reduce the safety factor to three and proceed with extreme care. A safety factor of three is the very least that can be justified, even when all possible precautions are taken. These include the exact determination of the in-water weight of the object; the accurate prediction of breakout force; the minimizing of weak points by use of new, unspliced line and the highest quality fittings; and the calculation and control of dynamic loads. Even if the lift is successful the line should be scrapped, its safe working load probably having been exceeded.

Table 1. Approximate weight constants of proportionality for various types of cables.

Type of Rope		Weight Constant of Proportionality (lbs/ft-in²)	
Material	Construction	In air, Ca	In Water, Cw
Steel Wire	6 × 24 Fiber Core, flattened strands	1.80	1.45
	6 × 37 Galvanized hawser	1.55	1.20
	6 × 24 (7 fiber core) Galvanized	1.40	1.05
	6 × 24 (steel core hoisting rope)	1.85	1.61
Nylon Rope	3-strand towline	0.29	0.028
	2-in-1 Samson	0.26	0.026
	Uniline	0.34	0.033
Kevlar Rope	2-in-1 Samson	0.38	0.057
	Uniline	0.39	0.058
	Phillystran 29	0.26	—
Polypropylene	2-in-1 Samson (nylon cover)	0.24	0
Polyester	2-in-1 Samson	0.32	—
	Uniline	0.39	—

Cable weight = (weight constant) × d^2
d = cable diameter in inches

Table 2. Approximate ultimate tensile strength constants of proportionality.

Material	Type of Rope Construction	Tensile Strength Constant of Proportionality (lbs/in^2) In air, Cua	In water, Cuw
Steel Wire	6 X 37 (steel core) galvanized steel rope	8.8×10^4	
	6 X 37 (steel core) galvanized monitor steel hawser	7.3×10^4	
	6 X 24 (7 fiber core) galvanized monitor steel rope	6.5×10^4	
	6 X 19 (steel core) hoisting rope	9.0×10^4	
Dacron Rope	Plain-lay (3-strand) heavy marine lay	1.85×10^4	1.85×10^4
Nylon Rope	Plain-lay (3-strand) regular lay	2.3×10^4	$2.1\text{-}2.4 \times 10^4$
	Plain-lay (3-strand) heavy marine lay	2.3×10^4	2.1×10^4
	2-in-1 Samson	2.9×10^4	2.55×10^4
	Uniline	4.0×10^4	
Kevlar Rope	2-in-1 Samson	8.7×10^4	
	Uniline	9.8×10^4	
	Phillystran 24	8.0×10^4	
Polyester	2-in-1 Samson	2.6×10^4	
	Uniline	4.0×10^4	
Polypropylene Rope	2-in-1 Samson (nylon cover)	2.6×10^4	2.5×10^4
	Plain-lay (3 strand) heavy marine lay	1.80×10^4	1.90×10^4
Manila Rope	Shroud-lay (4-strand) regular lay	1.00×10^4	
	Plain-lay (3-strand) regular lay	9.0×10^3	
Coir Rope	Plain-lay (3-strand) regular lay	2.5×10^3	0.23×10^4

Cable strength = (tensile strength constant) $\times d^2$
d = cable diameter in inches

Figures 2, 3 and 4 give the ultimate tensile strength of wire rope, nylon rope and Kevlar Uniline in various stock sizes over the range in depth down to 22,000 feet. Wire rope supports its own weight to 18,000 feet with a safety factor of three. Thus, it might be employed down to about 15,000 feet if all possible precautions are taken. Synthetic lines are almost independent of depth.

Greater depths can be achieved with steel rope, provided that suitable lengths of increasing sizes are spliced together. Appendix A gives an example of the calculations.

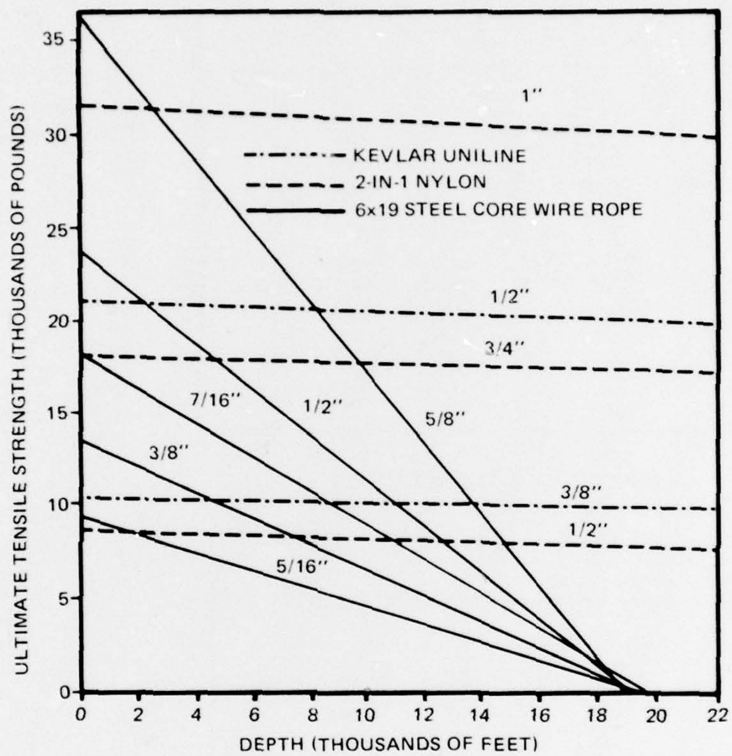

Figure 2. Ultimate tensile strength of various diameter lifting lines (safety factor: 3).

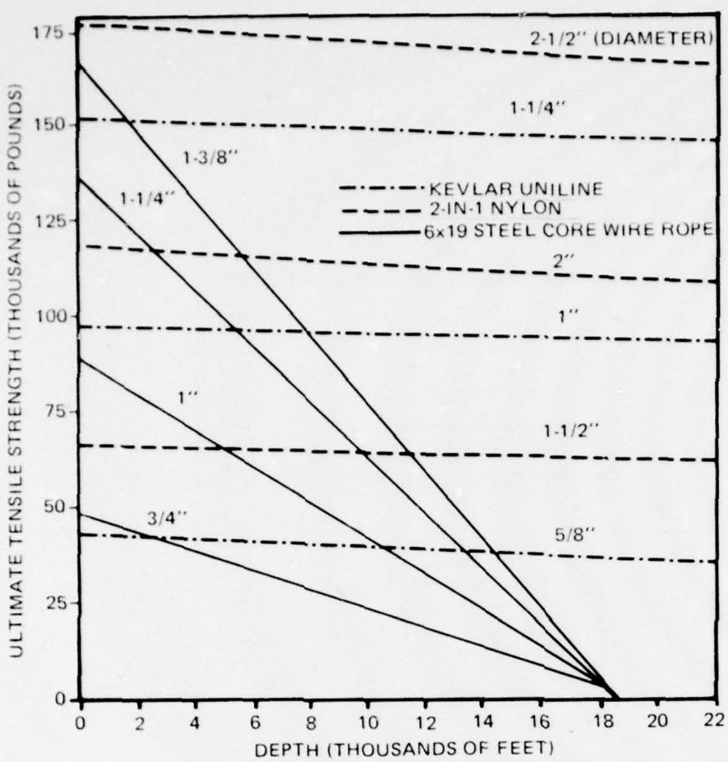

Figure 3. Ultimate tensile strength of various diameter lifting lines (safety factor: 3).

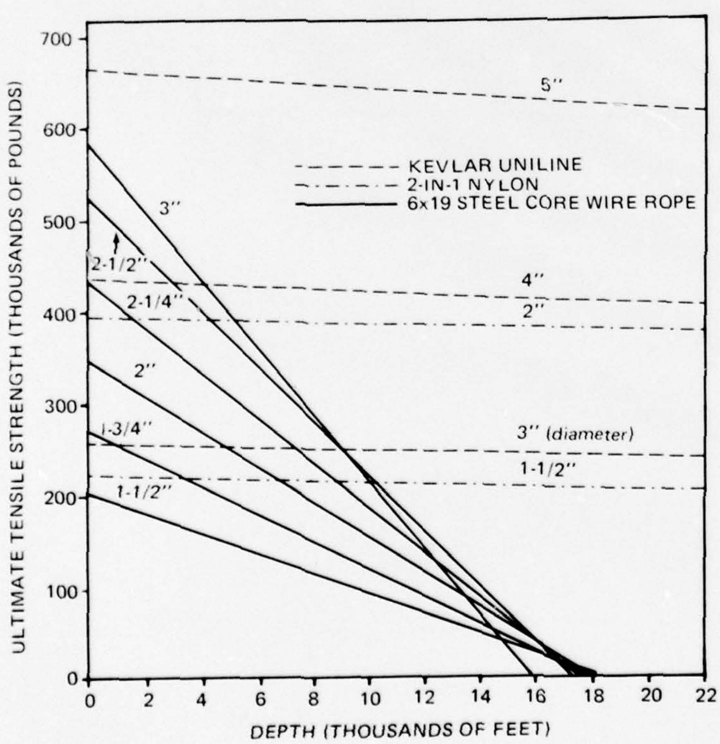

Figure 4. Ultimate tensile strength of various diameter lifting lines (safety factor: 3).

DYNAMIC LOADING

An accurate prediction of dynamic loading on a lift line is beyond the current state-of-the-art, but it is possible to understand the ingredients of the problem and to develop a partly numerical and partly intuitive feel for their effects. The ingredients of the problem are what happens at the top of the line, what happens to the line itself, and what happens to the load on the lower end.

Forcing Function

Problems originate at the upper end of the lift line, where forced vertical oscillation is imposed. This oscillation is predictable only as probability distributions of displacements, velocities, and accelerations.

The wave energy spectrum can be derived for specified weather conditions at sea. It is also possible to calculate a response operator that converts a wave motion spectrum to a vessel motion spectrum if vessel characteristics and heading are known. Approximations exist in every step of the calculations, but for reasonably fair weather the results will be sufficiently accurate for the mission. The geometry of the boom tip's location with respect to the vessel's center of mass then yields the motion spectrum of the upper end of the line. Appendix B is an example.

Figure 5 illustrates the statistical nature of ocean waves. It should not be construed to be a list of safety factors for operational use, but merely a table of odds. The highest wave in a week could occur at any time.

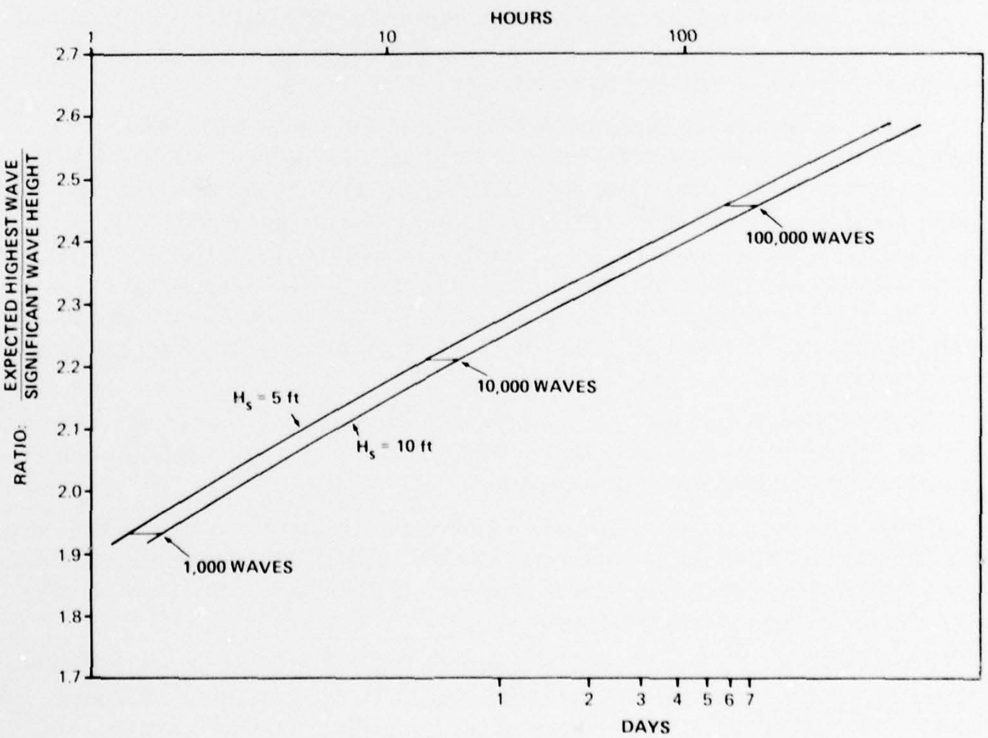

Figure 5. Expected highest wave: ratio to significant wave height.

Resonance

If an object on the end of a line is pulled down and then released, the extra stretch put into the line will make the object bob up and down briefly, like a weight hanging on the end of a spring. The frequency of this motion is called the fundamental resonant frequency, or natural frequency. The vertical motions of the boom tip send waves of alternating tension and relaxation propagating down the line in exactly the same manner that sound waves propagate through air as waves of alternating compression and expansion. Thus, the speed at which disturbances propagate up and down the line is known as the speed of sound for the line. If the frequency of the boom tip motion is the same or nearly the same as the natural frequency of the line, the line will transmit energy from the boom tip to the object at the lower end. The object will then oscillate up and down with increasing amplitude either until water resistance dissipates stored energy as fast as energy is added at the surface, or until the line parts.

The natural frequency depends on the length of the line, the thickness of the line, the type of material of which it is made, and the type of construction. The springiness of a line is affected, just as its strength is, by the type of construction: conventional rope laid, braided, or uniline (bundled parallel fibers). The most important difference, however, is in the type of material. Three types must be distinguished.

Steel cable has the stiffest elasticity, as well as the greatest strength for a given price. It has the disadvantage, however, that it is very heavy, even in water. A cable must carry its own weight as well as the weight of the object being lifted, and must be sized accordingly. All of this mass can resonate. For a deep lift with an ample safety factor, the mass of the cable is more significant than the mass of the object.

Kevlar cable, virtually as strong as a steel cable of equal diameter, has about half the stiffness in tension (modulus of elasticity) as a steel cable. Its weight in water is not a problem, even for a deep lift, but its mass is significant.

Figure 6 illustrates the elastic properties of steel and Kevlar cable. Both obey Hooke's Law, which states that in an elastic material, strain is proportional to stress. Within the elastic limits of each, stress is proportional to strain; tension-per-unit-of-cross-sectional-area is proportional to fractional elongation. This proportional constant is on the order of 20 million psi for steel cable, 12.5 million psi for a specially-manufactured Kevlar 49 cable such as the Remote Unmanned Work System (RUWS) primary tether, and 9.7 million psi for off-the-shelf Kevlar 29 cable. These numbers are only approximate, and must be expected to vary during the service life of a cable as well as from one manufacturer to another and from batch to batch.

Polyester uniline does not quite comply with Hooke's Law. The tensile modulus of elasticity gradually decreases from about 850,000 psi at no load to 400,000 psi at a load equal to half the ultimate tensile strength.

Nylon does not comply with Hooke's Law at all, whether rope-laid, braided, or uniline. Neither does rope-laid polyester (see figure 7). These lines stretch very easily at first, and then gradually stiffen as the load increases. This behavior has important consequences in predicting the natural frequency.

Figure 8 shows the natural periods of various types of lift lines, based on 20,000 feet of length. The independent variable is the safety factor: the ratio of breaking strength to the static weight of the object plus suspended line. For the synthetic lines, this safety factor is computed at the upper end. The steel cable is continuously tapered;

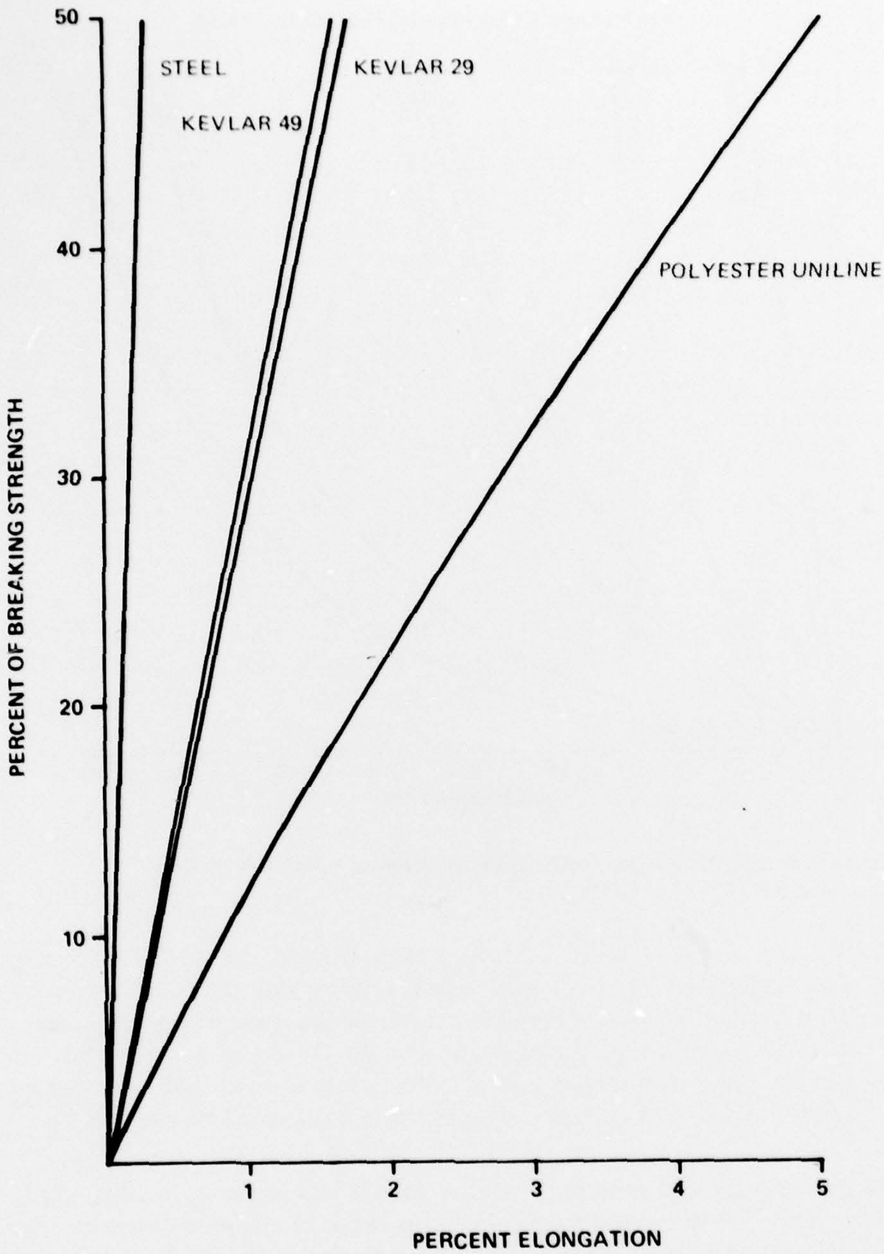

Figure 6. Percent elongation versus percent of breaking strength for elastic rope materials.

therefore, the safety factor is constant from top to bottom. Subsequent figures will show how the natural period varies with lengths of line from zero to 20,000 feet.

Nylon, of whatever form, performs very well in a 20,000-foot line. The natural frequency is very low, corresponding to periods of from 35 seconds to over one minute. The natural frequency depends on the method of construction, but the safety factor matters very little.

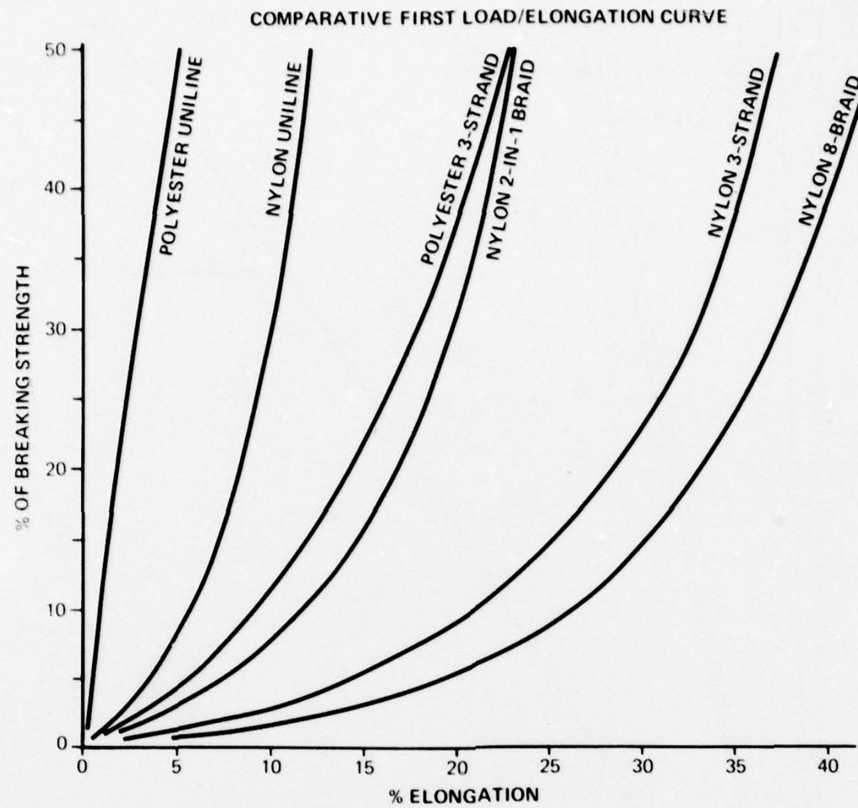

Figure 7. Percent elongation versus percent of breaking strength for various types of rope.

A 20,000-foot length of Kevlar cable may cause trouble. Its natural frequency is in the range of wave excitation. The frequency depends strongly on the safety factor, however. As the safety factor is increased, the natural frequency of the stronger cable will also increase. This behavior is characteristic of elastic systems. It should be noted, however, that reduction of the safety factor to reduce deep resonance is ineffective. A weaker cable will merely resonate at a shallower depth. This paradox is discussed by example in appendix C.

A 20,000-foot tapered or stepped steel cable may also become resonant. Such an extremely long cable has its natural frequency in the range of maximum wave energy for sea states 3 to 5. Initially, the curve parallels that for Kevlar; as the safety factor increases, so does the natural frequency. But the curve reverses direction at a safety factor of about five. Further increases in the safety factor result in a cable so massive that its natural frequency decreases.

Figure 9 is a graph of resonance in tapered steel cables of any length, showing the resonant period corresponding to the length for a variety of safety factors ranging from three to ten. (In figures 9 through 11, the safety factor is held constant for all lengths of

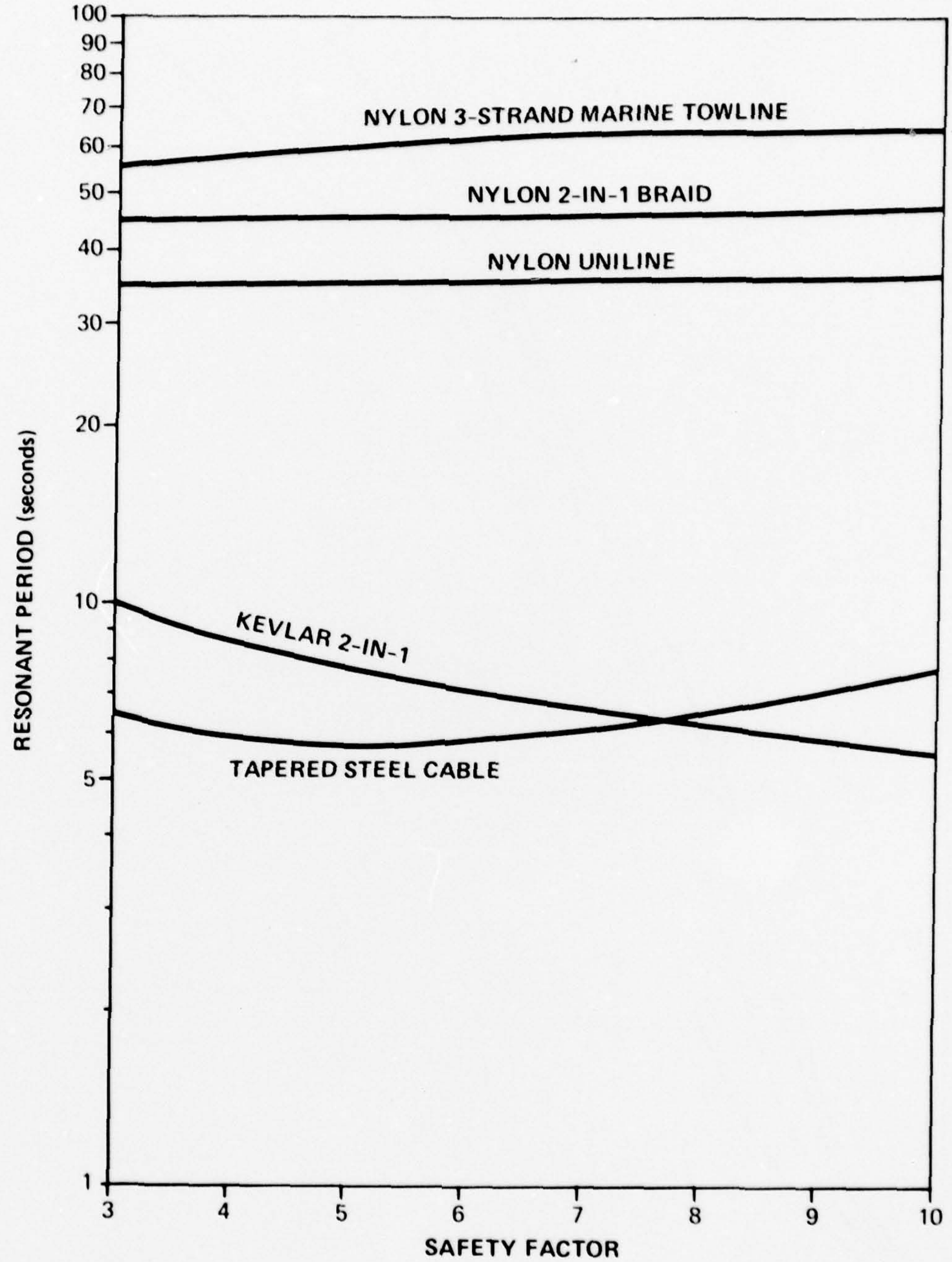

Figure 8. Natural period for 20,000 feet of line.

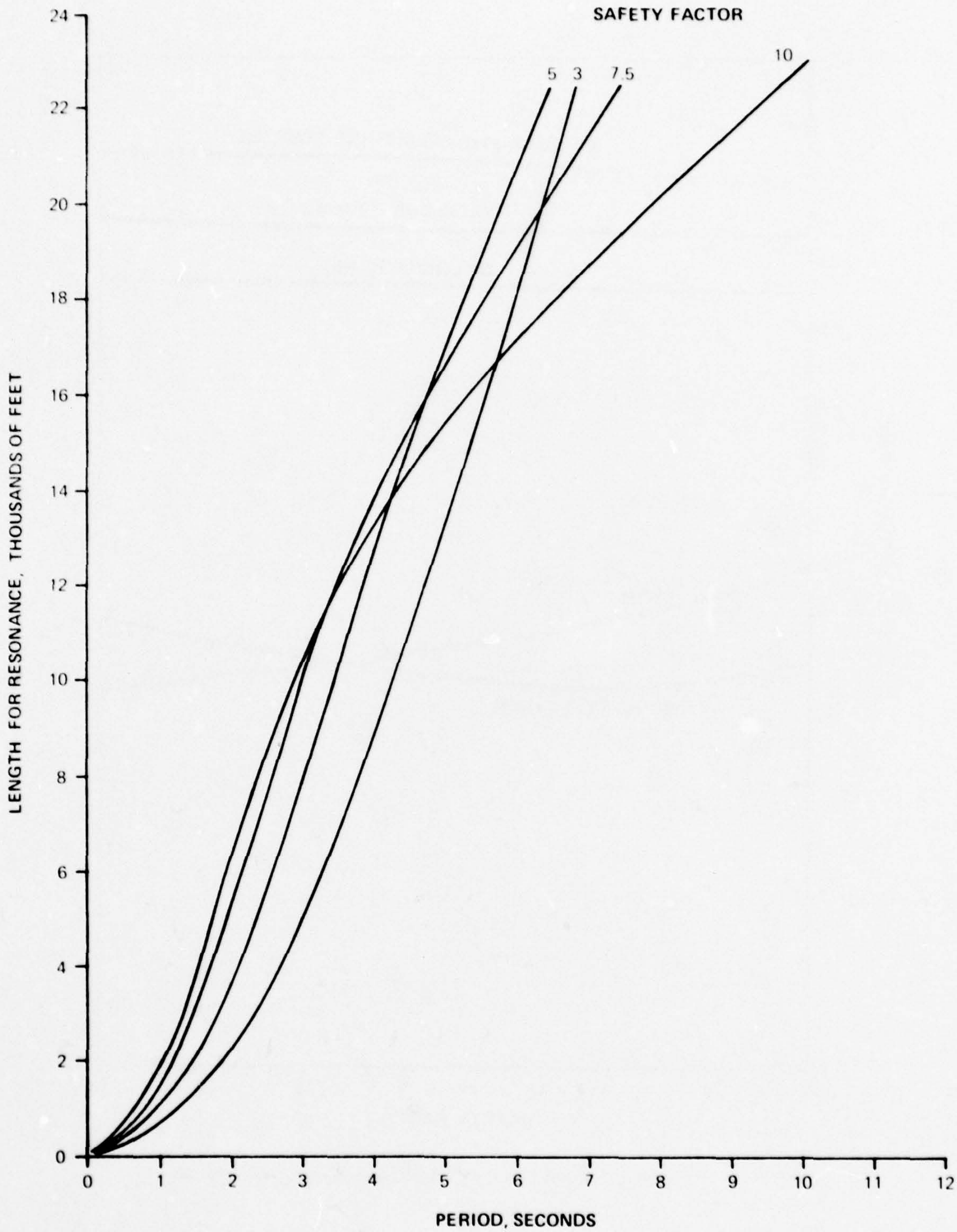

Figure 9. Resonant period for tapered steel cable.

line. As the length is increased, and the weight in water of the line increases, the weight of the object on the end is decreased by the same amount.)*

Figure 10 is a similar graph for Kevlar 2-in-1 cable. Kevlar is much more practical than steel cable for a deep lift. It is 27 times stronger in relation to its weight in water. But the natural frequency is a function of the safety factor as well as the length of the line. For lifts from 10,000 feet and below, the natural frequency corresponds to the frequency band in which most of the wave energy is concentrated for sea states 3 to 5. Only for lengths of 6,000 feet or less with a safety factor of 10 (and 2,000 feet with a safety factor of three) is the resonant period 3.0 seconds or less.

Finally, figure 11 presents comparable data for various forms of nylon: three-strand marine towline, 2-in-1, and uniline. Safety factors range from three to ten, but make almost no difference in the resonant period. As the weight and mass of the object vary, the modulus of elasticity of the line (and therefore the spring constant, which in this case is not a constant) varies almost in direct proportion. If it has been determined that the dangerous wave energy will be concentrated at periods of from 3.0 to 12.0 seconds, figure 11 indicates that safe lengths of nylon line are 1,250 feet and over for three-strand marine towline, 1,700 feet and over for 2-in-1, and 2,750 feet and over for uniline.

One conclusion that might be drawn is that a deep lift might best be made in two stages. The object might first be raised to a point near the surface with a synthetic line. The more elastic the line, the closer the object could be brought to the surface without danger.**

Dynamic Loading

The weight in water of the object, plus the weight in water of the line, is what determines the safety factor for the lift. However, the resonant frequency of the system depends on the dynamic mass of the object. This has three components:

- The mass of the object itself is proportional to its weight in air, not its weight in water.
- The object is flooded, and has a contained mass proportional to the volume of contained water.
- There is an added mass effect, external to the object.

As the object is accelerated, the water around it is also accelerated, making way forward of the object and closing in behind. An extra force is required, beyond the force required for the same acceleration in a vacuum. The effect is as if a certain body of water, called the added mass, were attached to the object and accelerating with it. The object seems more massive than it really is. If it is bobbing up and down on an elastic line, it

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*As an example of how this might be employed, suppose that after forecasting the weather and sea state it appears there will be no dangerous wave energy with periods of 3.0 seconds or less. Referring to figure 9, it may be seen that for lengths of 10,500 feet or less and a safety factor of 10, or 5,000 feet or less and a safety factor of three, the resonant period is 3.0 seconds or less.

**With nylon 2-in-1 line, a safe depth might be 2,500 feet. The load could then be transferred to a steel cable for the final stage of recovery. Note, however, that splicing steel end-to-end with nylon does not solve the problem, since the resonance will still occur in the nylon when its length reaches that noted in figure 11.

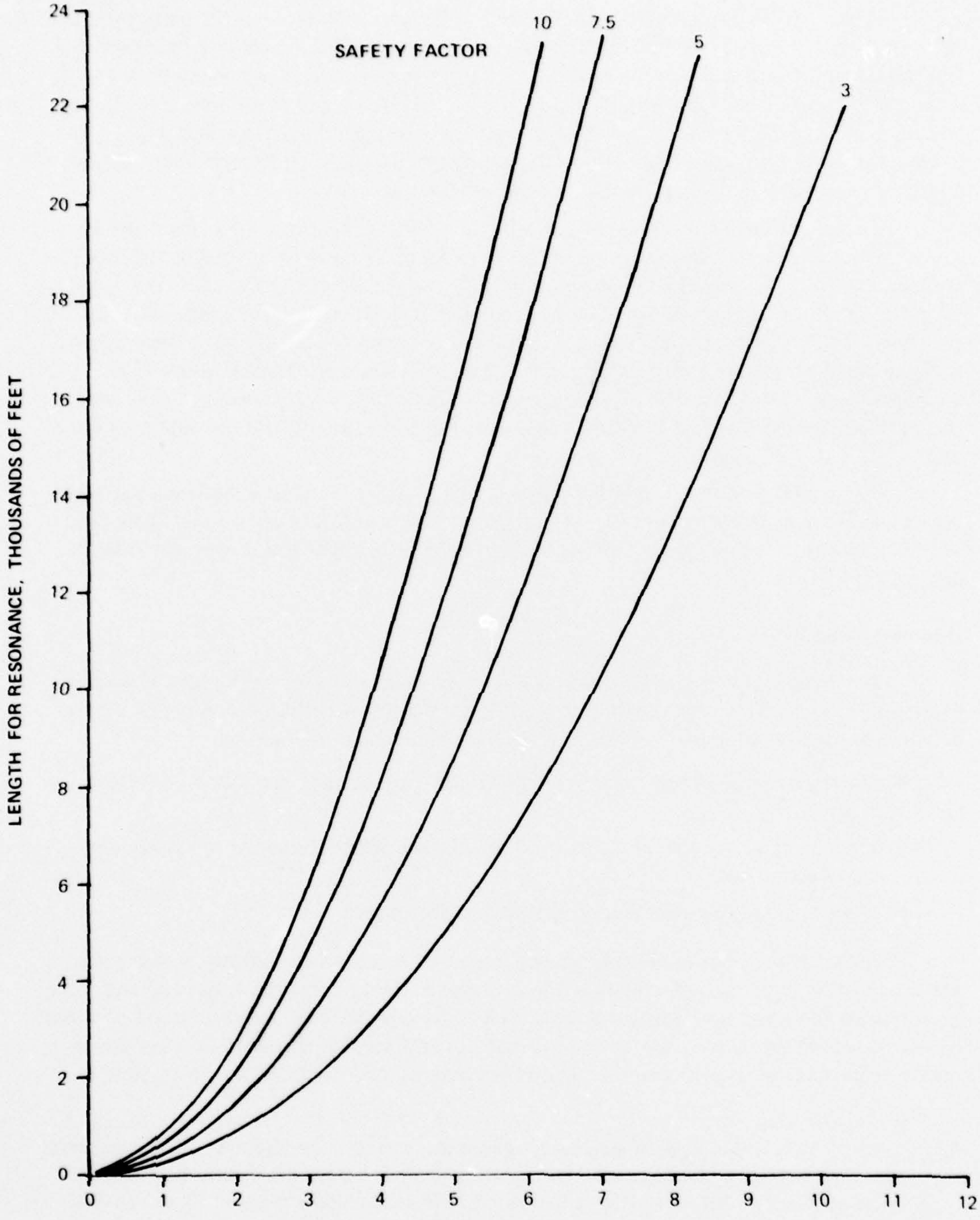


Figure 10. Resonant period for Kevlar 2-in-1.

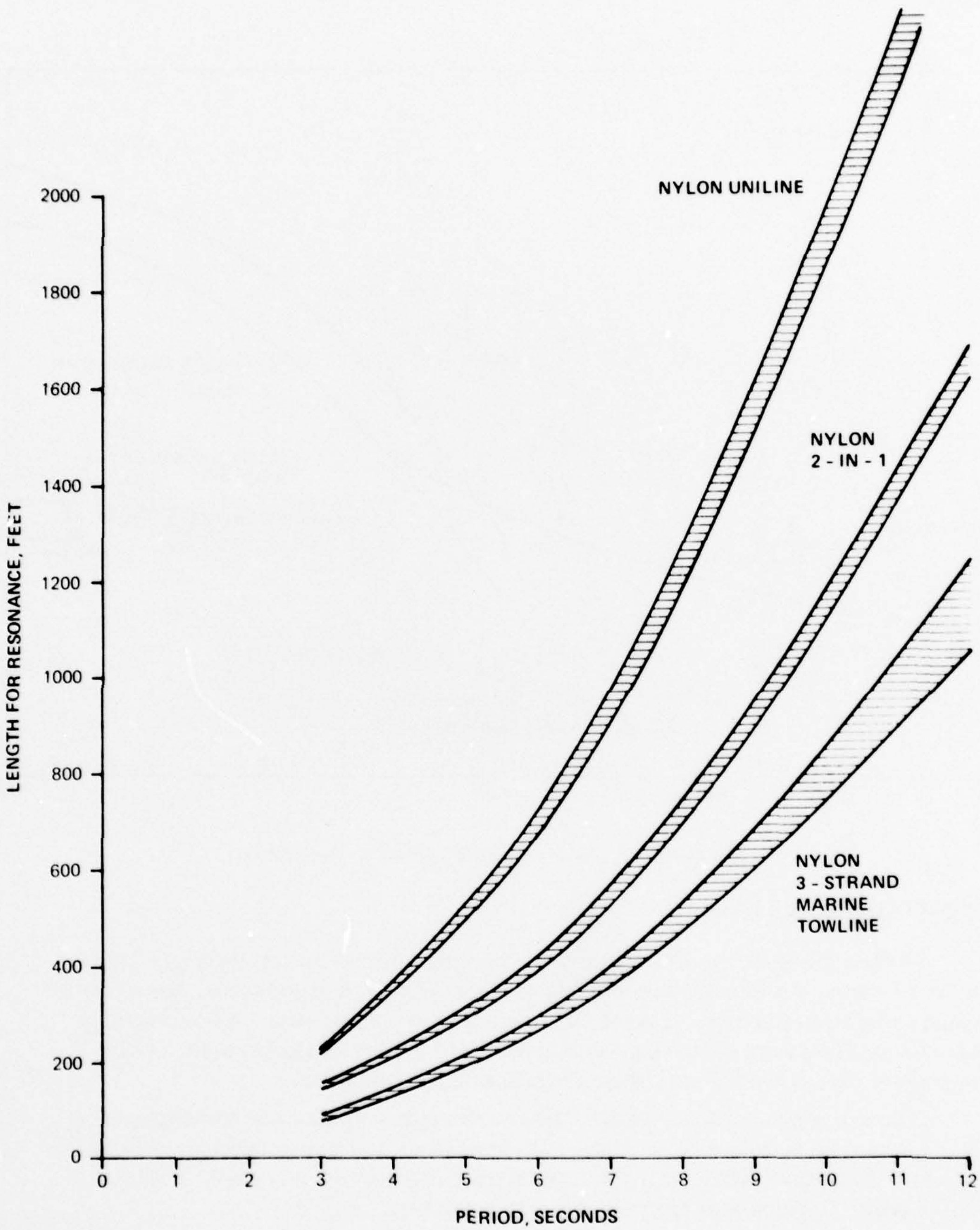


Figure 11. Resonant lengths of various types of nylon line (for safety factors from 3 to 10).

vibrates at a lower frequency than it would vibrate in air. An example of this is presented as appendix D. Figure 12 presents curves for the calculation of added mass coefficients and reference volumes.

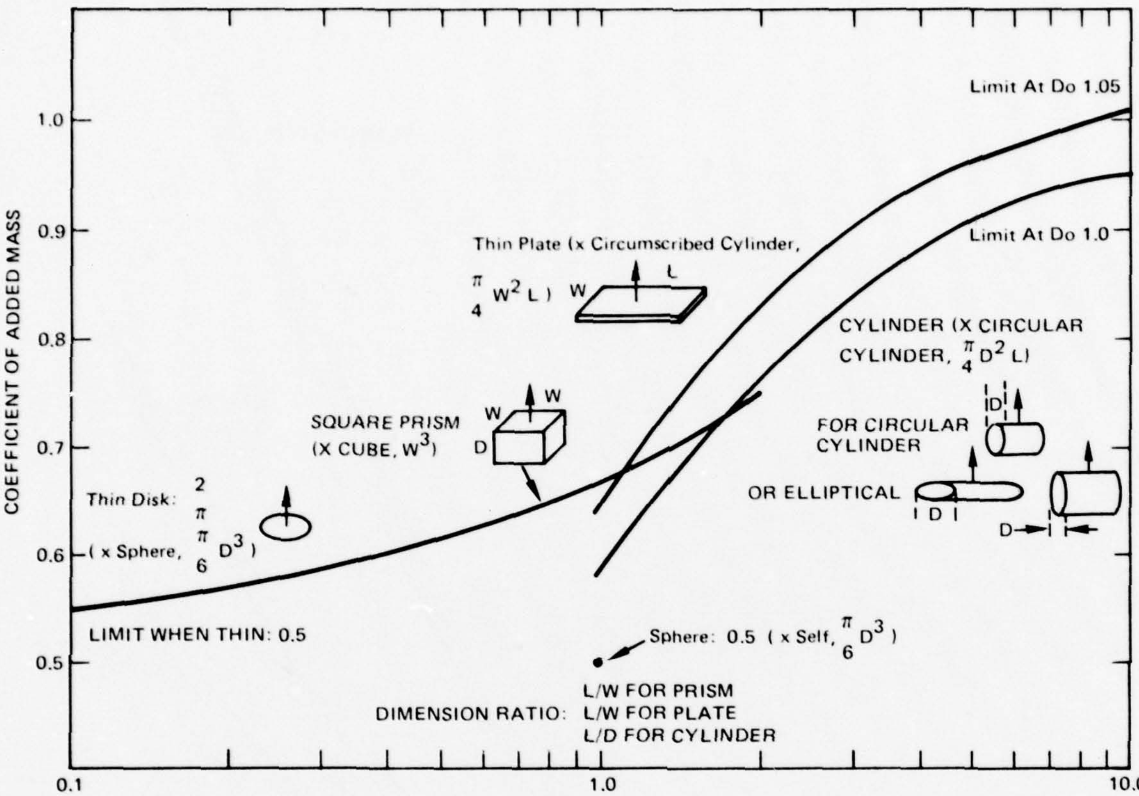


Figure 12. Added mass coefficients and reference volumes.

OTHER CONSIDERATIONS

Fatigue life is an important consideration, particularly for deep lifts requiring a number of hours. An initially ample safety factor may degrade rapidly with time, due to internal abrasion of the cable. Tests have shown that wet, three-strand nylon will only survive about 600 cycles of loading to 60 percent of initial breaking strength. If the average wave period is 6 seconds, there are 600 waves in one hour.

Allowance must also be made for loss of strength at splices and terminations. If properly done, these should be as strong as the line itself, but in practice they may be only about 80 percent efficient. Also, some synthetic lines, nylon in particular, lose between 5 and 20 percent of their dry strength when wet.

Another peculiarity is rotation, particularly of wire rope. This is really three problems in one. The tendency of wire rope to rotate is a problem in many applications, and special ropes are available which will exert a minimal torque on a suspended load. However, these ropes will still rotate if allowed to do so, and will lose a considerable amount of strength in the process. Tests on 1 1/4-inch diameter, 8 X 17 nonrotating wire

rope showed losses in strength of 25–30 percent when the load end was free to spin. In comparative tests, regular-lay, six-strand wire rope rotated much more when the end was free to turn, but its strength increased slightly.

Wire ropes must not be allowed to go slack after lifting or setting down a suspended load if the end of the rope is fixed from rotating. There is acute danger of a kink or a birdcage forming, either of which will significantly reduce the strength of the line. Synthetic lines with braided or bundled construction are specially manufactured to avoid problems of rotation and kinking.

One final caution must be observed when a safety factor is to be determined. Estimation of the static load itself is difficult, particularly if the object may be partly or fully flooded or filled with sediment.

A lifting line under tension has stretched and stored a great deal of energy. If it breaks, this energy is a hazard to men and equipment at the surface. Nylon and similar lines may more readily handle dynamic loads because of their vastly greater energy storage capacity. If they do break, however, the danger is increased in the same proportion.

When nylon and polyester lifting lines first came into general use, there was serious concern about the energy stored during breakout of the object from the sediment. It was feared that the object would be catapulted far from the bottom, that the line would go slack, and that the subsequent jerk as the line came taut again would part the line or tear loose the object. However, this has not been a major problem thus far.

Similarly, the dynamic loads resulting from the sudden starting and stopping of the winch are not generally a problem. A substantial effect occurs, however, if the line is near a resonant point.

Dynamic loading would be a much less serious problem if a constant-tension winch could be developed. Computer studies at the Naval Civil Engineering Laboratory (reference 4) indicate that a winch which would accommodate line oscillations with a displacement of plus or minus nine feet, a velocity of plus or minus seven feet per second, and an acceleration of plus or minus 1/4 g would be adequate for use in sea state four on ocean-going tugs and salvage ships. These values are consistent with those in the table in appendix B.

Costs

Table 3 offers a comparison of the costs of various types of lifting lines in 1978 dollars. The cost indices can be used with a fair degree of accuracy in determining the cost of lines ranging from 1/2 inch in diameter up to the maximum diameter produced by a given manufacturer. For instance, if 20,000 feet of a 2-in-1 nylon line are desired with a breaking strength of 120,000 pounds, the cost of the line would be, approximately:

$$3.3(10^{-5})\,(20{,}000)\,(120{,}000) \approx \$79{,}200$$

Samson quotes their 2-inch-diameter line (breaking strength equals 131,000 pounds) at $3.87 per foot, or $77,400.

An important point to remember is that although the cost index of wire is low compared to the synthetics, the weight of the suspended wire must be added to the weight of the load being lifted to determine the required breaking strength.

Table 3. Cost comparison for various types of lifting line.

Type of Line	Cost Index* $/ft/pound breaking strength
6 × 19 IWRC Wire Rope	1.6×10^{-5}
Nylon Uniline	3.0×10^{-5}
2-in-1 Nylon	3.3×10^{-5}
Kevlar Uniline	3.4×10^{-5}
Phillystran	8.2×10^{-5}
2-in-1 Kevlar	8.7×10^{-5}

*Based upon average cost figures for line in the 1-to-2 inch diameter range. A variation of 20-30 percent can be expected on a given quote.

DRILL STRINGS

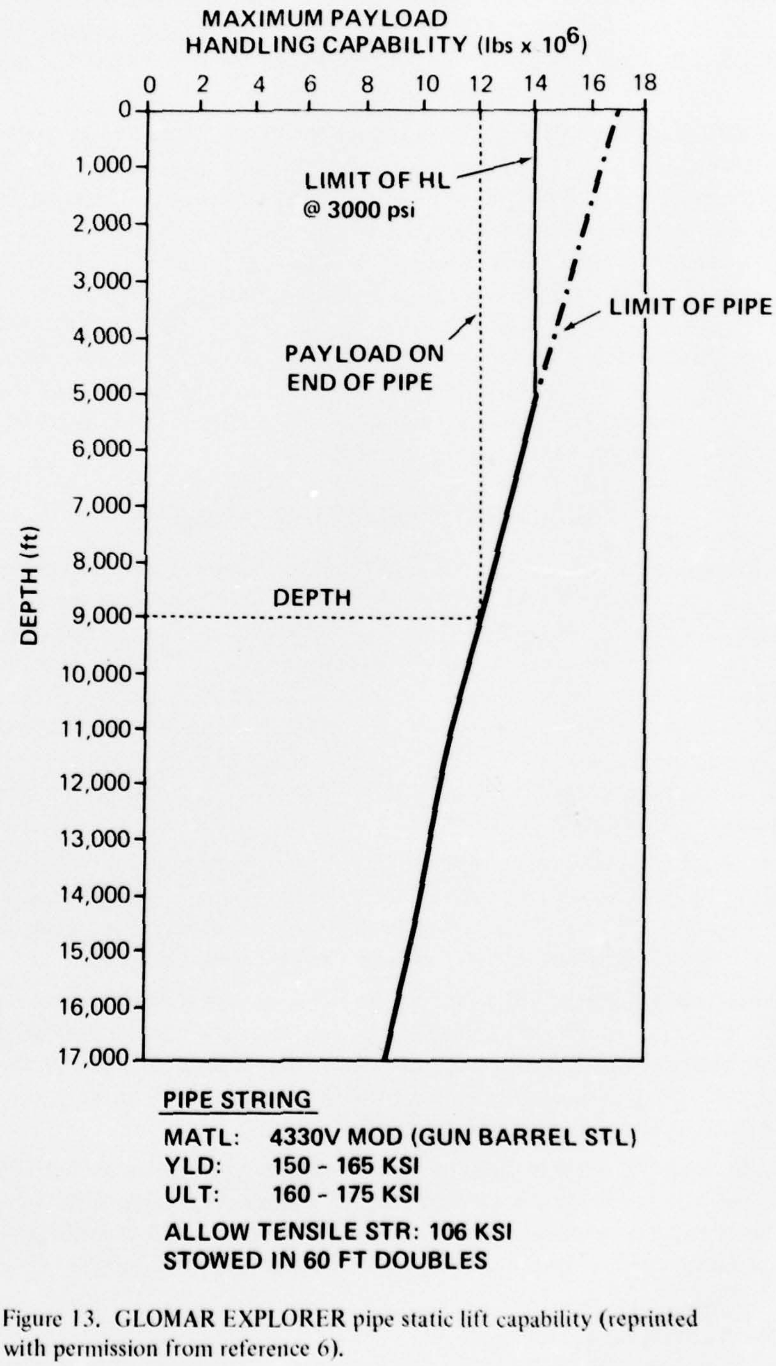
The use of drill strings to perform heavy lifts in the ocean has been considered since the first ocean-going rigs appeared in the late fifties. In a report issued in 1969 (reference 5), NCEL (now CEL) concluded from a comprehensive analysis of ship and pipe dynamics, and of various cost and operational considerations, that the most feasible system for lifting and raising loads of 100 and 600 tons in 6,000 feet of water was through the use of drill strings mounted on special floating platforms or drill ships. More recently, the GLOMAR EXPLORER was given the design capability of raising over 4,000 tons from depths exceeding 17,000 feet. Although drill ships, drill barges, and semi-submersibles could all be used for direct lift, only drill ships have been investigated to any extent for the present study because of their inherent mobility and the ability of some to position themselves dynamically. Dynamic stationkeeping has proved to be more practical than deep-ocean mooring for a short-term operation. Only one semi-submersible drill rig, the SEDCO 709, has a dynamic positioning capability. No drill barges do.

LOAD CAPABILITY

With the exception of the GLOMAR EXPLORER, the rated load lifting capability of most drill ships and semi-submersible drill platforms of interest for heavy lift operations is in the range of 500 to 665 short tons. This is the gross lifting capability on deck. The weight of the drill string must be subtracted to obtain the net lifting capability as a function of depth. In addition, the weight of the connection to the object must be subtracted in determining the maximum weight of the object to be lifted. Because of their size and stability, the lifting capacity of the larger drill ships is not seriously altered in weather as bad as sea state five. Also, shock absorbers (called bumper subs) have been provided to reduce the effects of vessel motions on the suspended pipe strings, and rubber pipe protectors are used by the GLOMAR CHALLENGER to dampen vibrations in the drill string.

For deep operations, the net lift capability of drill ships ranges from 50 tons at 20,000 feet for the ALCOA SEAPROBE to 4,250 tons at 17,000 feet for the GLOMAR EXPLORER. ALCOA SEAPROBE has a 500-ton derrick but is limited by a draw works

capacity of 250 tons. She has a net capability of 233 tons in 2,000 feet of water, 200 tons in 6,000 feet of water, and 50 tons in 20,000 feet of water. At present, however, she is outfitted with only enough pipe to reach 9,550 feet. The GLOMAR EXPLORER uses a stepped drill string of six different pipe diameters ranging from 12.75 to 15.5 inches for a maximum length of 17,363 feet. Figure 13 indicates how the net lifting capability of GLOMAR EXPLORER varies with depth.



PIPE STRING

MATL: 4330V MOD (GUN BARREL STL)
YLD: 150 - 165 KSI
ULT: 160 - 175 KSI
ALLOW TENSILE STR: 106 KSI
STOWED IN 60 FT DOUBLES

Figure 13. GLOMAR EXPLORER pipe static lift capability (reprinted with permission from reference 6).

The net lift capability of intermediate drill ships must be calculated. A sample calculation is included as appendix E.

COST OF OPERATION

As a general rule, the day rate of a drilling vessel is approximately $1,200 to $1,300 per million dollars of capital investment in the vessel. For ALCOA SEAPROBE, this is $6,500 to $7,000 per day. GLOMAR EXPLORER, reputed to have cost about $65 million, should rent for about $80,000 per day. Global Marine ships of the CHALLENGER class have a day rate of about $20,000; the new PACIFIC class, about $50,000.

Most drill ships have a speed capability of about 10 knots. Thus, they can be expected to cruise about 200 to 250 miles per day and to require four to five days to get on site for an operation 1,000 miles from their initial location. Once on site, most drill ships can lower pipe at 40 to 60 feet per minute, which means that about seven hours would be required to lower 20,000 feet of pipe string. The GLOMAR EXPLORER only lowers pipe at six feet per minute; thus, a minimum of two days is required to lower 17,000 feet of pipe. Generally, any operation that goes well requires about two weeks or more of on-site ship time. The cost of renting a ship alone for a two-week operation would range from about $100,000 for the ALCOA SEAPROBE to $1.2 million for the GLOMAR EXPLORER. The total cost of the operation will be even greater due to the cost of additional personnel and equipment required.

FIXED AND VARIABLE BUOYANCY

The literature on fixed and variable buoyancy includes a confusing variety of measures of effectiveness by which to evaluate a float. The basic measure is the specific density (or specific gravity) of the float, relative to seawater. The specific buoyancy is the difference between the specific density of seawater itself (of unity) and the specific density of the float. The effective density of a float is its weight per unit volume, while its effective buoyancy is its buoyancy per unit volume. The displacement factor is the inverse of the specific density, or the displacement of the float in pounds per pound of float. Finally, the buoyancy factor is the net buoyancy of the float per pound of float. These measures are summarized in table 4.

Materials of historical importance include wood, with an effective buoyancy of about 20 pounds per cubic foot for construction lumber and about 48 pounds per cubic foot for balsa wood. The useful operating depth of wood is limited to about 700 feet. Wood becomes waterlogged very rapidly at such depths, even if painted.

The buoyancy material employed in the first manned descents was aviation gasoline. The effective buoyancy of a system of this type is 10 to 20 pounds per cubic foot. Greater buoyancy is obtained with the more volatile and hazardous hydrocarbons. Such an undersea blimp is limited in maneuverability and seaworthiness, resulting in a sacrifice of effective buoyancy.

Lithium metal, the lightest of the solid elements (specific density 0.515 relative to seawater) has been used, well-potted to prevent chemical reaction with water. The effective buoyancy, after precautions have been taken, is 17 to 25 pounds per cubic foot. The syntactic foam blocks now available are safer and less costly.

Table 4. Buoyancy factors.

Symbol	Name	Formula
ρ_s	Specific Density	$\rho_s = \dfrac{\text{Float Weight in Air}}{\text{Weight of Water Displaced}}$
B_s	Specific Buoyancy	$B_s = 1 - \rho_s$
ρ_{eff}	Effective Density	$\rho_{eff} = 64\ \rho_s$
B_{eff}	Effective Buoyancy	$B_{eff} = 64\ B_s$
F_D	Displacement Factor	$F_D = \dfrac{1}{P_s}$
F_B	Buoyancy Factor	$F_B = F_D - 1$

RIGID SHELLS

A rigid shell containing air at atmospheric pressure can provide buoyancy if it is sufficiently light and strong. Such a shell is the most convenient form of float, since it can also serve as a housing for equipment. The strongest shape for such a shell is the sphere. Figure 14 is a plot of the collapse depth of a spherical shell as a function of the specific density and the buoyancy factor for a variety of materials.

Figure 14 applies to near-perfect spheres. It can be shown that the collapse pressure of a theoretically perfect sphere would be 1.44 times that for a near-perfect sphere. Even the strength implied by figure 14, while attainable in practice, cannot be guaranteed by quality assurance procedures. Therefore, the calculations have been run out to more than twice the deepest depth of the ocean so that a safety factor of two or more may be applied.

The collapse pressure of a sphere does not depend on the thickness or the diameter, but only on the ratio of the two. It is therefore a function of the specific density and independent of size. Each of the curves for a specific material in figure 14 has a knee, although in some cases the knee is off the graph. For thinner spheres (those below the knee), buckling is the pertinent mode of failure, and the modulus of elasticity (denoted by E) is the controlling material property. For thicker spheres (those above the knee of the curve), compressive yielding is the failure mode, and the compressive yield strength (σ) is the controlling property.

Rib-stiffened cylinders are much more convenient than spheres, although not as strong. Figure 15 is a plot similar to figure 14, but for cylinders, replotted from reference 7. Again, an ample safety factor must be applied.

Characteristics of commercially available spheres are given in table 5. A comparison of table 5 with figure 14 shows that the Corning Pyrex spheres are rated with a safety factor of 2.5 with respect to nearly-perfect spheres and 3.6 with respect to perfect spheres.

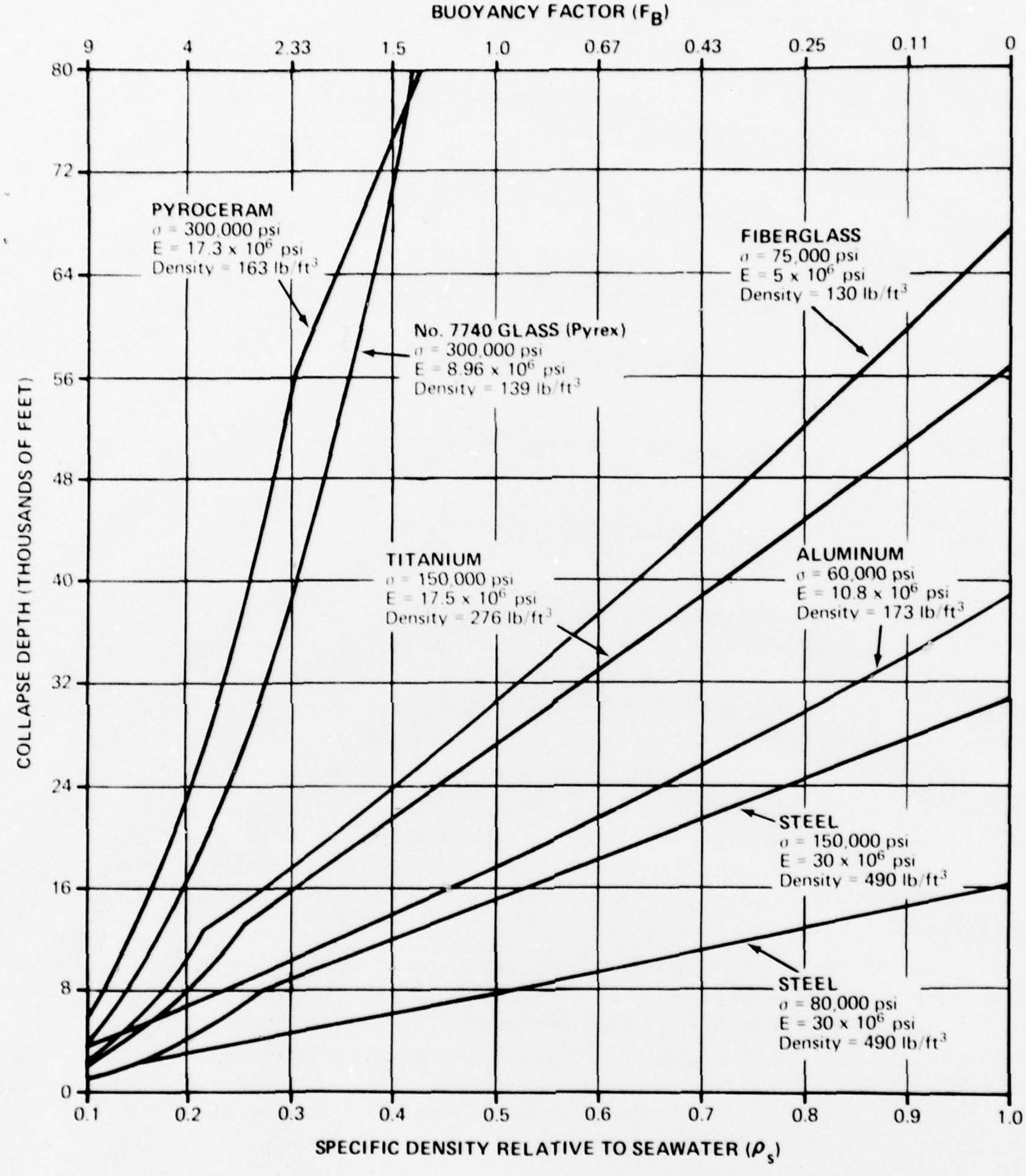


Figure 14. Estimated collapse depth versus specific density and buoyancy factor for near-perfect spheres.

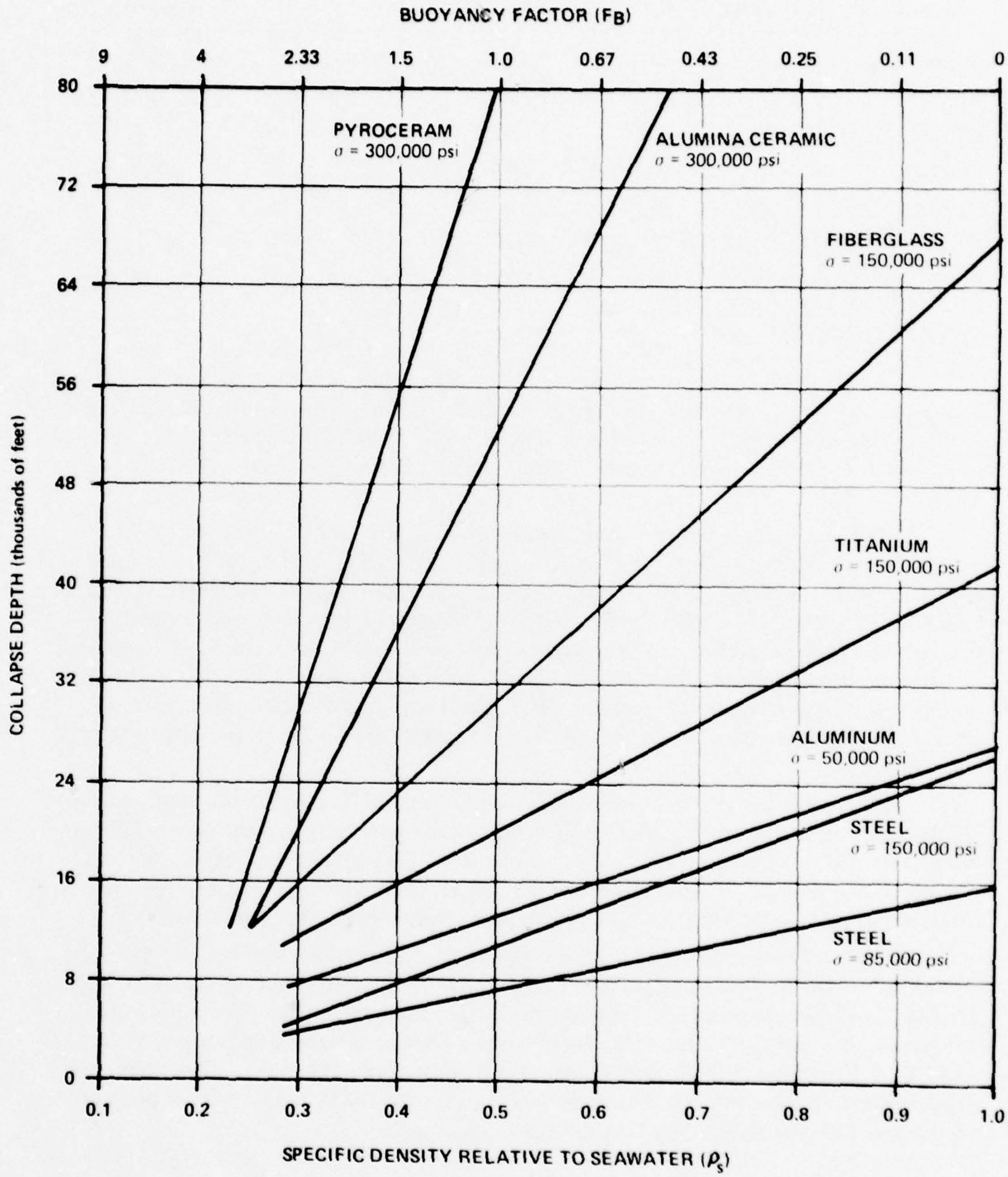


Figure 15. Estimated collapse depth versus specific density and buoyancy factor for rib-stiffened cylinders.

Table 5. Characteristics of commercial spheres.

Manufacturer	Diameter (Inches)	Material	Displacement Factor FD	Weight (lb)	Buoyancy (lb)	Operating Depth (ft.)
Benthos	10	Glass	2.25	9	11	22,000
Benthos	13	Glass	2.47	17	25	22,000
Corning	16	Glass	2.65	30	49	23,000
Benthos	17	Glass	2.44	39	56	22,000
Corning	20	Glass	3.01	52	104	20,000
Inter Ocean	31	Steel	3.25	178	400	2,500
Inter Ocean	32	Steel	2.70	235	400	5,000

SYNTACTIC FOAM

Syntactic foam is an increasingly attractive form of buoyancy. This is resin filled with microscopic glass spheres. The term "syntactic" is an exaggeration; it is related to "syntax" and implies an ordered arrangement of the spheres, like atoms in a crystal lattice. In reality, the spheres are not of uniform size and are randomly packed. The term, then, expresses an ideal.

Syntactic foam is a very safe material with which to work. It does not in fact collapse at its so-called collapse depth, well beyond its rated service depth. Its collapse depth is merely the depth at which deterioration – surface cracking and waterlogging – is significant after a few hours of exposure. The manufacturer's rated service depth is the depth at which deterioration is essentially zero on a time scale of hundreds of hours. The true limiting depth of a particular foam can only be determined by the user, based on the type and duration of the mission. For example, on a brief, one-time, down-and-up mission, syntactic foam could perhaps be taken to or beyond its rated collapse depth with no danger.

Figure 16 displays the Emerson & Cuming company's rated service depth versus density for four grades and 17 weights of syntactic foam. The first grade developed for full service, EG, weighs just over 34 pounds per cubic foot when rated for 20,000 feet. The superior grade, EL, weighs just under 32 pounds for the same condition. EX weighs 29 pounds, and TG, the best grade, 27 pounds per cubic foot at 20,000 feet service depth.

Figure 17 presents the same data rearranged as a function of price in dollars per pound of buoyancy. The upper pair of curves refers to the two standard grades of foam, EG and EL. The curious knee in both curves is the result of the manufacturer's pricing policy. In both grades, the least expensive foam is the one weighing 34 pounds per cubic foot. Both lighter and heavier weights are significantly more expensive. The other curve, in the lower right corner of the figure, shows the two premium grades of foam progressing both in cost and service depth as the density increases.

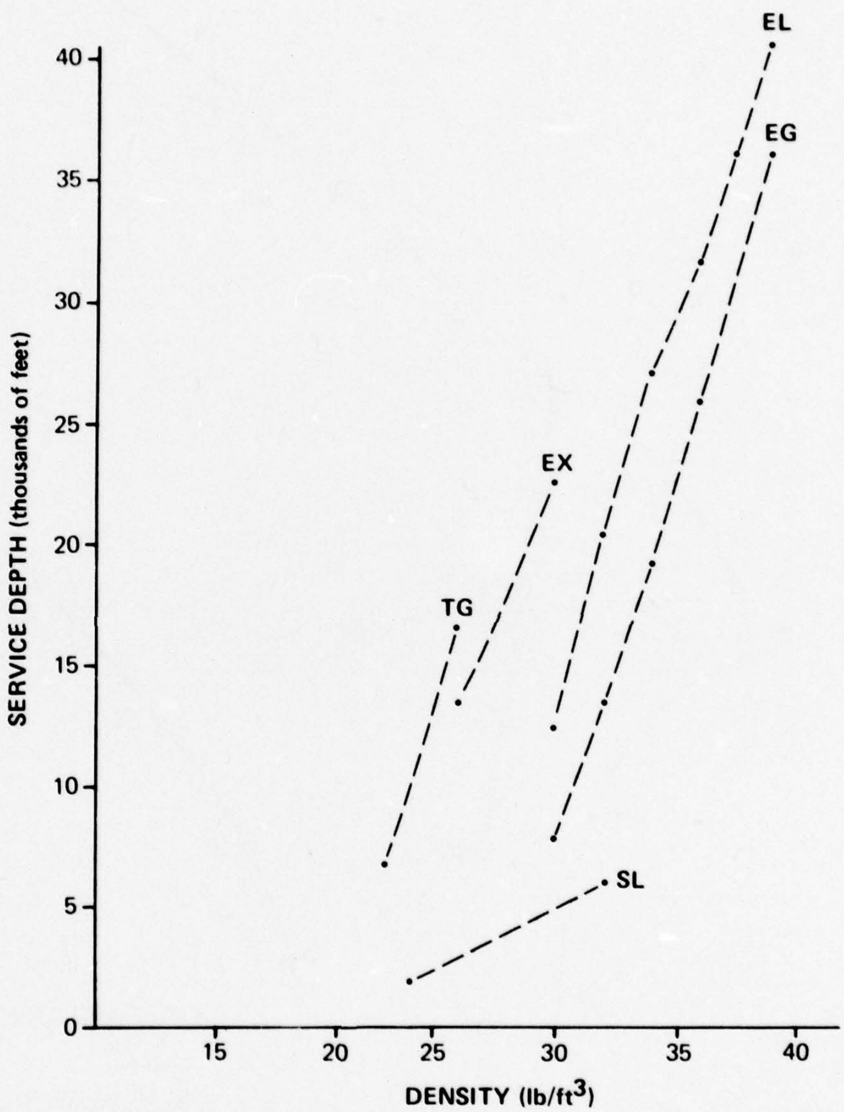


Figure 16. Service depth versus density for various grades of syntactic foam.

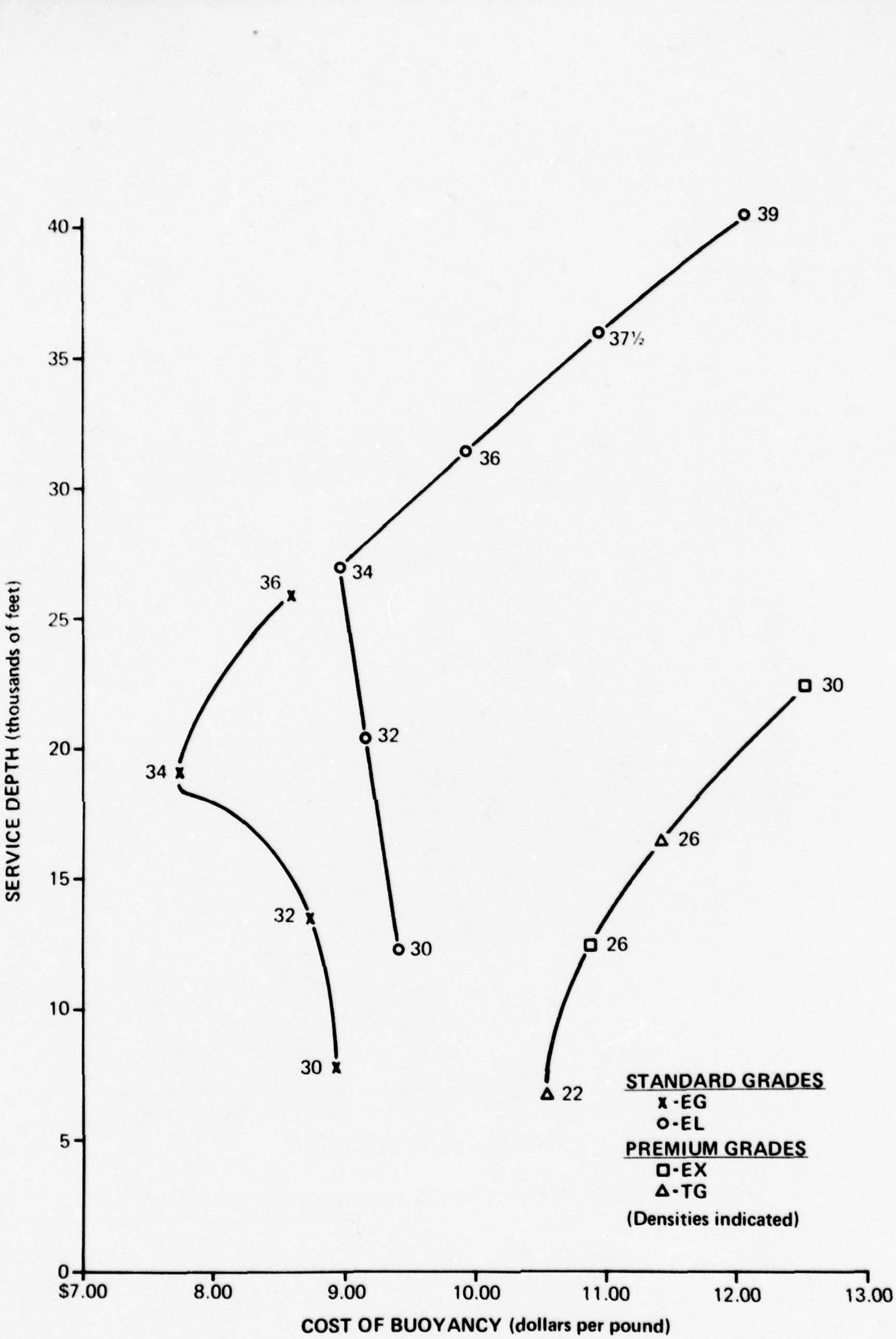


Figure 17. Service depth versus cost per pound of buoyancy for various weights of syntactic foam.

VARIABLE BUOYANCY

Variable buoyancy can be obtained in four ways: by combining ballast with fixed buoyancy; by flooding and dewatering a pressure-resistant shell; by blowing a rigid pontoon or ballast tank with a compressed gas; and by inflating a flexible pontoon.

Any of the fixed buoyancy devices described in the preceding section can be converted into a variable buoyancy system by the addition of droppable ballast. Iron shot carried in a hopper has proved to be economical, efficient (Ps=7.66 for iron, relative to seawater), and failsafe. An electromagnet wound around the bottom opening of the hopper will, as long as it is energized, prevent the shot from falling out.

Alternately, a pressure-resistant shell in the form of a sphere (figure 14) or rib-stiffened cylinder (figure 15) may be used as a ballast tank. The tank is filled with a specified amount of water to make it neutrally buoyant at the surface. The ballast water is then pumped out at depth to produce the required buoyancy. This is illustrated in appendix F.

Pontoons are lightweight, unpressurized gas tanks or gas bags containing air or some other gas at ambient pressure. As the pontoon rises and the ambient pressure drops, excess gas is bled off through a low-pressure relief valve or a short standpipe below the pontoon.

Pontoons may be either flexible or rigid. Small pontoons, up to 10 tons of buoyancy, can be fabricated of rubberized fabric for ease of storage and handling. Larger pontoons are more conveniently fabricated of steel plate than of cloth. In the future, imaginative use might be made of other rigid materials such as reinforced plastic, or even reinforced concrete.

Table 6 compares the advantages and disadvantages of rigid shell pontoons.

Table 6. Advantages and disadvantages of rigid pontoons.

Advantages	Disadvantages
Ease of fabrication	Cumbersome to store and handle
Flexibility of shape	Some materials need corrosion protection
Low cost	Sensitive to impact
Ancillary equipment can be easily attached	Heavier than flexible shell pontoons
Pontoon may be compartmentalized	Will not withstand negative pressure
Pontoon volume does not vary with pressure	
Easily rigged	
Easily repaired or modified at sea	
Impervious to gas permeation	
High abrasion and puncture resistance	
Large range of pressures available (with accompanying weight increases)	

In the final analysis, however, it will be necessary to examine the choice of a flexible or rigid pontoon from a systems point of view. In a deep-ocean recovery system, it may not be convenient that the pontoon be a simple bag or tank with a single lift point and a single point of attachment. It may be desirable that the pontoon incorporate a large mechanism to solidly grapple the object being recovered. It may also be advantageous that the pontoon incorporate a complex assembly of gas generating equipment with associated control mechanisms, batteries, telemetry equipment, pingers and beacons. The principal advantage of a rigid shell pontoon is that all these related devices can be incorporated into the structure. While the thin walls of the pontoon will be susceptible to damage during rough water deployment, the main disadvantages of a rigid pontoon will probably be that its size and weight preclude airlift. For example, the Large Object Salvage System (LOSS) pontoon has a net capacity of 100 tons and a dry weight of 80 tons.

On the other hand, the complex mechanisms required for a deep-ocean recovery operation might all be incorporated into a large and remotely-controlled vehicle. Such a vehicle, however, may require various amounts of auxiliary buoyancy for lifting large objects such as an aircraft. A flexible, air-capable pontoon would be a necessary adjunct to such a system, particularly if the vehicle itself were designed for air transport.

Table 7 (reference 8) provides the parameters of a range of sizes of flexible pontoon. The pontoons are similar to the teardrop-shaped, 10-ton and 1.5-ton units fabricated from Neoprene-coated Kevlar fabric for the Naval Ocean Systems Center by the B. F. Goodrich Company in fiscal years 1975 and 1976. The values in the table are rounded, but dimensions and stresses are accurate within plus or minus two percent; weight within plus or minus five percent.

Table 7 shows that a flexible pontoon providing 50 long tons of buoyancy at the surface could be constructed from Kevlar fabric. The height would be 19 feet, the diameter just under 15 feet, and the total assembled weight (with a safety factor of five), 1,000 pounds: all values well within reason.

Problems have been experienced during recovery operations in which a pontoon and suspended object have been allowed to come to the surface with too much speed or momentum. The pontoon has broached, vented an excess of air, and sunk back to the bottom. The problems here are complex, although open to analysis and solution through a study of system dynamics. These unfortunate occurrences, which have involved both hard and soft pontoons, do not invalidate the concept of pontoons or preclude their use in salvage work. If a pontoon is allowed to broach, the pressure differential inside must be sufficient to keep the pontoon filled with air as it settles again into the water. A pontoon that is merely open at the bottom or has a standpipe a foot or two in length does not develop a safe pressure differential. Such a pontoon is safe in a carefully controlled ascent, but not otherwise. On the other hand, a soft pontoon with an adjustable relief valve has been tested with perfect safety under extreme conditions (reference 9).

Table 7. Lift bag parameters.

Lift Capacity (Long Tons)	Height In.	Diameter In.	Bead Dia In.	Fabric Loading lb./in. max. Longtitudinal	Hoop	Bag Weight lbs.	Total Assy Wt. lbs.	Ratio Lift Cap. To Wt.
5	107	81	12	350	110	80	120	93
10	136	102	15	540	155	140	220	109
25	182	137	21	975	270	295	500	112
50	228	171	26	1570	415	570	1000	112

BUOYANCY GASES

The preceding section discussed hard and soft pontoons without mentioning how they might be inflated. Buoyancy gases fall into two categories: those that are presently available, and those which might be available with further development. The available gases are three: air, nitrogen, and the mixture of hydrogen and nitrogen obtained by the catalytic decomposition of hydrazine. At shallow depths, compressed air is the traditional choice. It also has the highest specific density, as shown in figure 18, reaching half that of seawater at 17,000 feet. Even at shallower depths, nitrogen and hydrazine offer definite advantages.

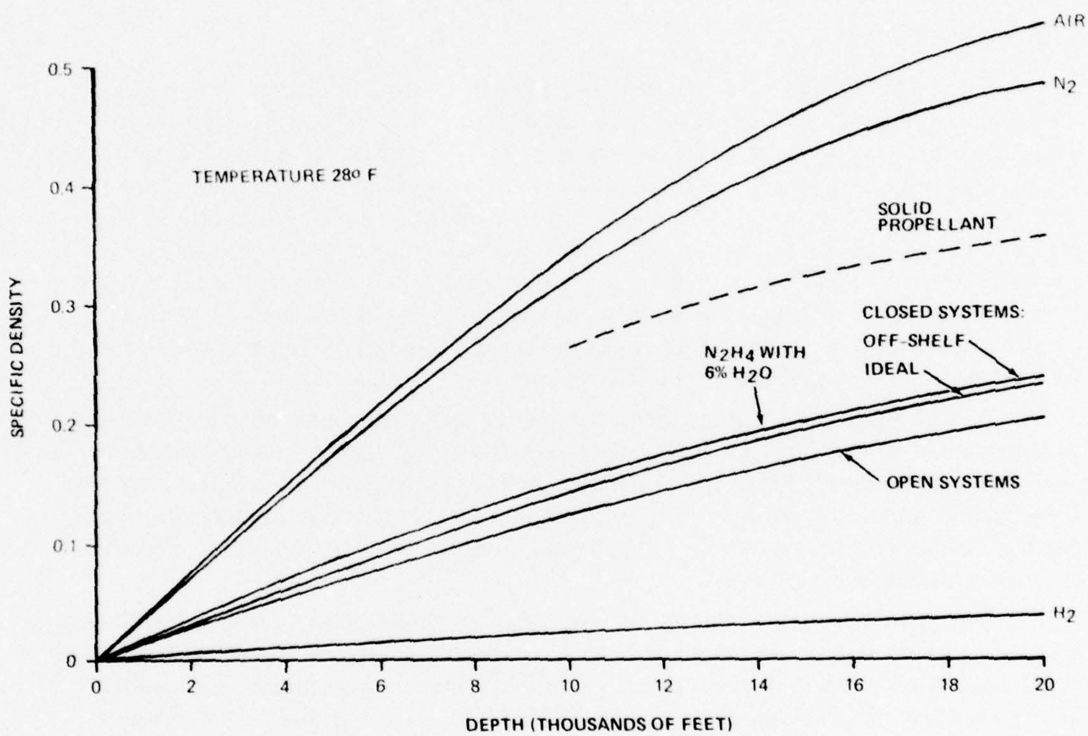


Figure 18. Specific density of buoyancy gases.

There are also three buoyancy generation mechanisms in a laboratory status. Hydrogen is the ideal buoyancy gas because it has the lowest possible specific density. Hydrogen could be generated by the reaction of lithium hydride with seawater. It could also be generated by a supercorroding alloy of magnesium and iron. Finally, Olin Corporation has developed a potentially convenient, nonexplosive, solid monopropellant rocket fuel, N–28. Its mixture of combustion products is more dense than that produced by hydrazine, but lighter than pure nitrogen.

COMPRESSED AIR

Compressed air, supplied from the surface, is the original buoyancy source and is still the most common buoyancy gas in present use. Salvage ships carry large air compressors, air banks, hoses, and fittings. These systems are designed for deployment by

divers, and are thus limited to about 850 feet. However, commercial air compressors delivering 2,500 psig and capable of delivering air to a depth of 5,000 feet are manufactured in very large sizes. For pressures above 2,500 psig, only small pumps and hoses of less than one-inch diameter are available.

The major advantages of a compressed air system are cost and convenience. Compressed air is practically free compared to alternative buoyancy gases. All complicated equipment is located at the surface. There is no significant penalty in efficiency; for depths of 5,000 feet and less the specific density of air is less than 0.2. The large, long, and heavy air hose may nevertheless present problems that outweigh the advantages of compressed air.

LIQUID NITROGEN

The Large Object Salvage System (LOSS) employs liquid nitrogen to provide buoyancy. The liquid nitrogen is contained in pressurized dewars inside the pontoon. These are double-walled tanks with an evacuated space between the walls which provides the thermal insulation to prevent the liquid nitrogen from boiling away before the pontoon is attached to the target. The dewars have a working depth of 1,000 feet. After attachment, the liquid nitrogen is forced through a heat exchanger, where it is warmed by seawater from -320° F to ambient temperature. The liquid nitrogen has a density of 52.5 pounds per cubic foot. It becomes a gas with a density of less than 2.5 pounds per cubic foot. Four long tons of nitrogen expand to fill the pontoon and provide 100 long tons of net buoyancy at a depth of 1,000 feet.

As buoyancy gases and cryogenic liquids, air and nitrogen are similar. The choice is not between air and nitrogen, but compressed air and liquefied gas. There would be no perceptible difference if LOSS were operated with liquid air rather than liquid nitrogen. Commercial liquid nitrogen is an inexpensive by-product of the manufacture of oxygen, but if it had to be manufactured on site at sea it would be simpler and just as effective to manufacture liquid air instead.

The main difficulty in employing cryogenic gas at depths greater than 1,000 feet is in the provision of an adequately insulated container. A vacuum barrier of reasonable weight would be crushed at such depths. So would an insulating blanket, for insulation of any kind is a series of minute air spaces which are highly susceptible to crushing.

HYDRAZINE

Although discovered a century ago, hydrazine (N_2H_4) was a laboratory curiosity until World War II. Difficult to prepare from ammonia as a weak aqueous solution, and even more difficult to obtain as the anhydride, it was so far from practical use that its physical and chemical properties had not been well measured. The Allied occupation forces were surprised, then, to discover German railroad tank cars of hydrazine hydrate ($N_2H_4 \cdot H_2O$). With hydrogen peroxide as the oxidizer, and mixed with alcohol, hydrazine hydrate was the fuel used in the rocket-powered ME-163 and Natter interceptor aircraft. Hydrazine has been an essential part of the U.S. space program, used as a monopropellant in the small attitude control rockets on spacecraft.

A hydrazine-fueled buoyancy gas generator was developed in conjunction with the LOSS program (reference 10). This generator can produce enough gas to displace 200 long tons of seawater at a depth of 1,000 feet. Even at this relatively shallow depth, a hydrazine gas generator has significant operational advantages as compared

with an air hose or a cryogenic gas supply Hydrazine gas generators are under active industrial consideration in connection with undersea oil production at depths of several thousand feet. At deeper depths, hydrazine is even more attractive in comparison with air and nitrogen. As shown in figure 18, the specific density of the gas produced at 20,000 feet is in the range of 0.2 to 0.24, depending on the design of the system.

Hydrazine is analogous to hydrogen peroxide. Ammonia, the parent substance, is a highly polar solvent, as is water, and occupies the same position in the nitrogen family as water does in the oxygen family. Hydrazine is related to ammonia in the same way that hydrogen peroxide is related to water.

In the presence of a suitable catalyst, hydrazine decomposes spontaneously into nitrogen and ammonia. A great deal of heat is liberated. Some of the ammonia then decomposes into nitrogen and hydrogen in the endothermic reaction that absorbs part of the heat from the first reaction:

$$3N_2H_4 \rightarrow N_2 + 4NH_3 + 144{,}300 \text{ BTU}$$

$$4NH_3 \rightarrow 2N_2 + 6H_2 - 79{,}200 \text{ BTU}$$

The first reaction goes to completion, but the second never does. It is customary to let X represent the reaction efficiency (the efficiency with which the ammonia dissociates. For buoyancy applications, X should be as near to unity as possible. Figure 19 gives the gas output in standard cubic feet per pound of hydrazine as a function of X. As X can approach or reach 80 percent in practice, a hydrazine gas generator can be expected to produce about 29 standard cubic feet per pound.

The temperature of the gas is inversely proportional to X, since the dissociation reaction is endothermic. Even with X equal to 80 percent, however, the reaction temperature is reduced only to 1,400°F, as shown in figure 20.

Figure 20 also shows that the molecular weight of the gas decreases as X increases. This is because the amount of hydrogen increases with X much faster than the amount of nitrogen, as shown in figure 21.

Hydrazine is a clear, thin liquid with many physical properties similar to water (see table 8). In particular, its density is equal to that of seawater. Thus, a bladder of hydrazine is neutrally buoyant in the ocean.

Hydrazine need not be a hazard aboard ship, if the appropriate safety precautions are understood and observed as they are for other potent substances carried at sea. Hydrazine does not deteriorate when stored properly, and there is no risk that it will decompose spontaneously and explosively. It cannot be set off by shock or friction. In short, it is safe in a way that most explosives are not. Since it is a powerful reducing agent, however, special measures must be taken. If it is allowed to come into contact with rust, an explosion may result. In storage, it should be kept covered with a blanket of nitrogen to displace air. It causes a caustic burn if it comes in contact with the skin. Also, it can be absorbed through the skin with damage to the liver. The permissible concentration of fumes is one part per million. All-day exposure to uncontrolled hydrazine fumes, even at levels too weak for detection of the characteristic ammonia odor, may cause painful inflammation of the eyes and temporary blindness.

Pure hydrazine is liable to freeze at the low temperature and high pressure of the deep ocean, and must therefore be mixed with a small percentage of water, which serves as an antifreeze. The freezing point of hydrazine is 34.25°F at one atmosphere of

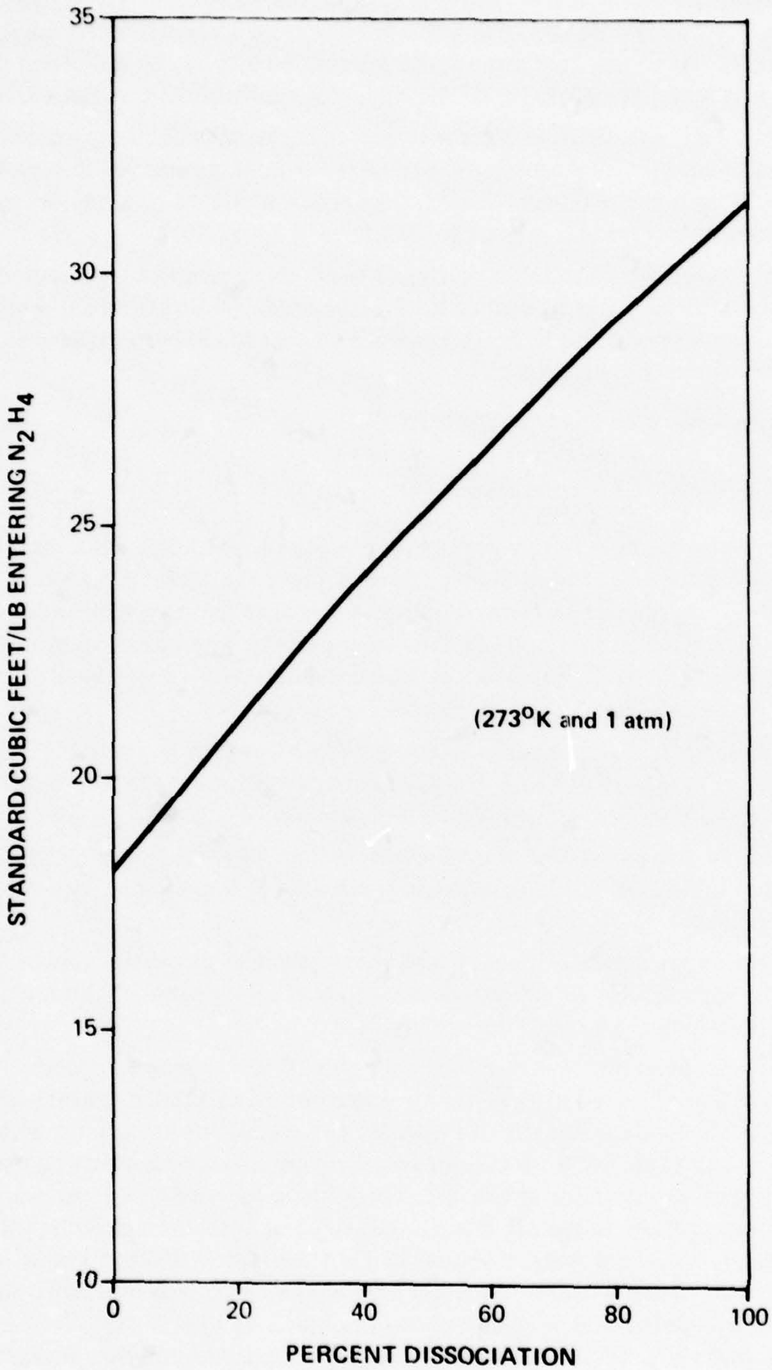


Figure 19. Buoyancy gas output per pound of hydrazine (reprinted with permission from reference 11).

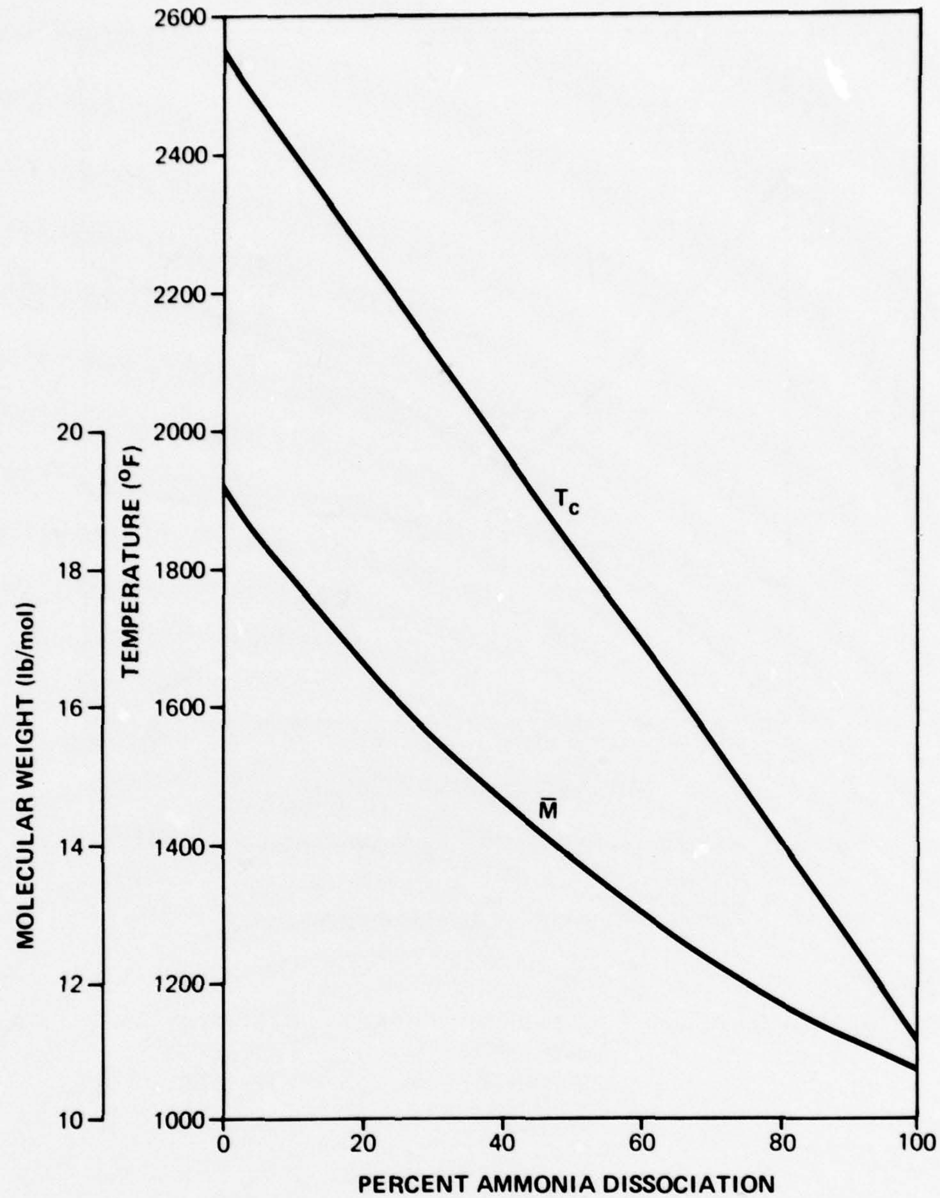


Figure 20. Reaction temperature and molecular weight of gas (reprinted with permission from reference 11).

pressure, but increases with depth to 41°F at 20,000 feet. This elevation of the freezing point with pressure is related thermodynamically to the way that hydrazine contracts while freezing.

As water freezes, it works against the ambient pressure. This work depresses the freezing point (see figure 22). Normal substances behave in reverse, as shown in figure 23. They contract as they change from liquid to solid. When they make this change under pressure, the pressure works on them, and this absorption of energy raises the freezing point. A small percentage of water mixed into hydrazine depresses the freezing point significantly. Figure 24 gives the freezing point of the blend as a function of both the

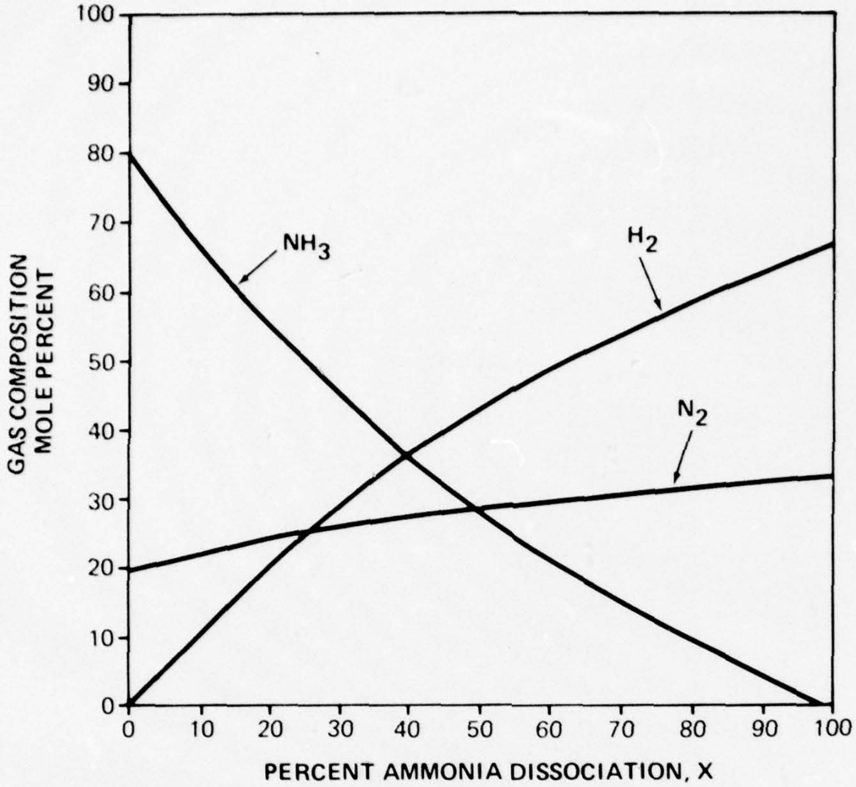


Figure 21. Gas composition (reprinted with permission from reference 11).

Table 8. Properties of anhydrous hydrazine.

Molecular Weight	32.04	
Density, liquid (g/cc; lbs/gal)	1.0253 (0°C) 1.0040 (25°C) 0.9801 (50°C)	8.557 (32°F) 8.379 (77°F) 8.179 (122°F)
Density, solid	1.146 (-5°C)	9.564 (23°)
Boiling Point	113.5°C	236.3°F
Melting Point	1.25°C	34.25°F
Heat Capacity (liquid) (cal/mole°C; BTU/lb°F)	23.62 (25°C)	0.737 (77°F)
Heat of Fusion	3025 cal/mole	170 BTU/lb
Explosive Limits (in air, 1 atm)	4.7% lower 100% Upper	
Flash and Fire Point	52°C	125.6°F
Fire Hazard	Moderate, when exposed to heat or flare	
Explosion Hazard	Severe, when exposed to heat, flame or oxidizing agents	
Fire Fighting	Water, mist, foam, CO_2, dry chemical or carbon tetrachloride	
Availability	99.5% hydrazine used as rocket fuel; available in 55 gallon (440 pound) drums and tank cars	
Price	\$3.40 per pound in 440 pound drums, Government price for mil-spec rocket fuel as of 1 April 1978	

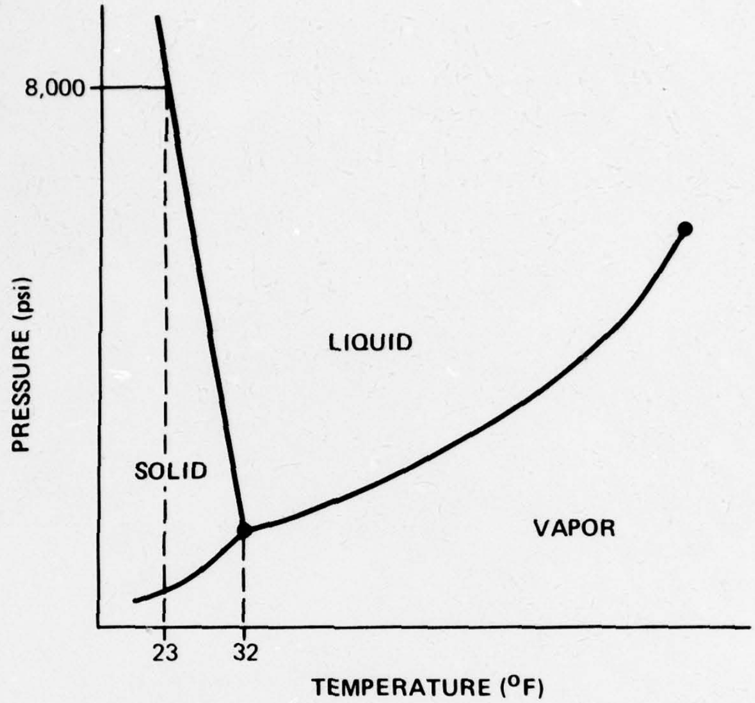


Figure 22. Phase diagram for water (reprinted from reference 9).

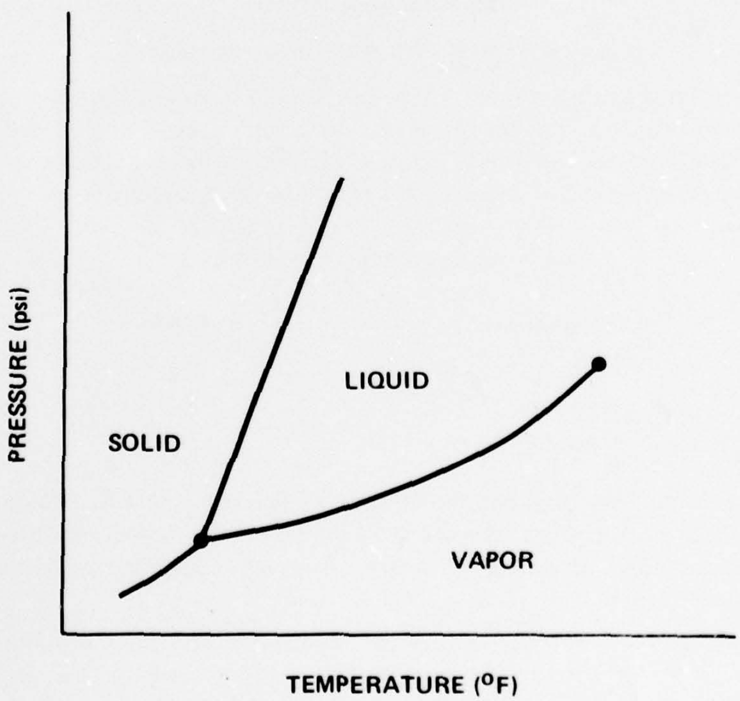


Figure 23. Phase diagram for a substance that contracts on freezing (reprinted from reference 9).

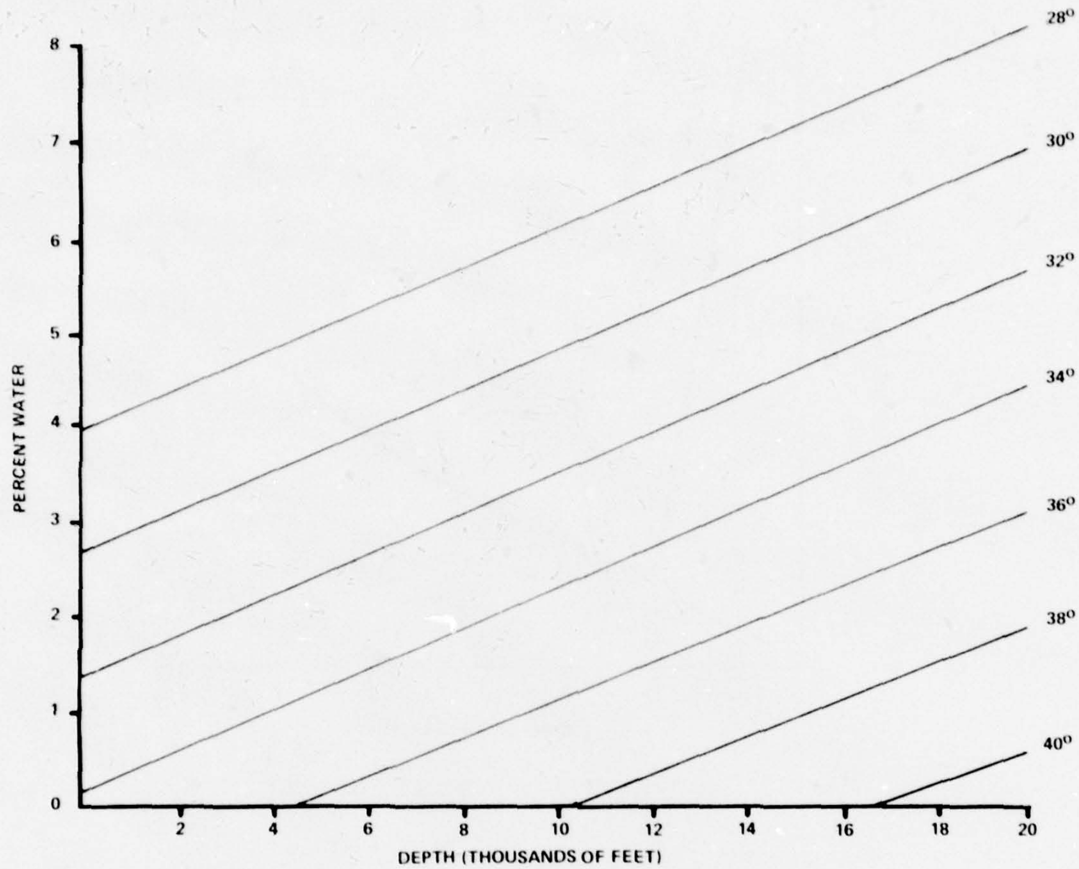


Figure 24. Freezing point of hydrazine-water blends.

fraction of water and the ocean depth. There will already be one-half of one percent water in the hydrazine as supplied. Rocket fuel is not chemically pure; it may be assumed to have the impurities indicated in table 9. Some of the following calculations are based on a 6-percent water blend, which is adequate for most buoyancy applications.

Table 9. Monopropellant grade hydrazine.

Hydrazine	98.97%
Water	0.52%
Aniline	0.37%
Ammonia	0.14%

A hydrazine buoyancy system is very simple. It consists of a fuel bladder with a pump or other arrangement to provide about 50 psi over ambient pressure; the reactor, a very small steel cylinder containing the catalyst; a length of steel tubing to cool the gas; various valves; and the pontoon.

Figure 18 shows the specific density for open and closed buoyancy systems. In a closed system, all the reaction products go into the pontoon and fill the available space. In addition to the nitrogen and hydrogen, there are the undissociated ammonia and the water. In an open system, the ammonia and water are eliminated. If the end of the gas tube is underwater it is an open system; the ammonia and water will be passed off into

the ocean as the gas bubbles into the pontoon. If the tube vents inside the pontoon, but the ammonia and water are blown out the bottom as the pontoon fills, it is still an open system.

The open system performs better, and the specific density is not perceptibly affected by either the dissociation efficiency or the water. In practice, these only affect the fuel tankage that must be provided.*

There has been a recent revolution in knowledge concerning the fractional dissociation of ammonia achievable in a hydrazine reactor. The original work at the Naval Weapons Center in 1972 (reference 13) indicated that the dissociation efficiency X increased with the logarithm of the reactor length (which, however, was severely limited by the pressure drop), but decreased with the logarithm of the ambient pressure. This curve of X against depth is shown in figure 25. The curve seemed logical because it paralleled the theoretical curve based on thermodynamic equilibrium between the ammonia and the dissociated gases, also shown in figure 25. An increase in pressure is expected to shift the equilibrium point toward the liquid phase, since the liquid occupies less volume.

Later work at the Civil Engineering Laboratory in 1976 (reference 14) was unable to verify this decrease in X with increasing pressure. There was even some indication that it might go the other way.

Most recently, Rocket Research Corporation, in unpublished work, has determined that X does indeed increase with ambient pressure. That is, it approaches the equilibrium curve asymptotically, and only begins to turn down at a depth of 15,000 to 20,000 feet when very close to the equilibrium value for that depth.

The original assumption was that a buoyancy system would require a custom-designed reactor with an efficiency of 0.75 to 0.8 at the surface and 0.55 to 0.6 at 20,000 feet. It is now apparent that an off-the-shelf rocket motor can be used. These are designed to yield X=0.4 in space, to maximize the specific impulse, but when used in the ocean they yield closer to X=0.8. The use of a six-percent water blend should lower performance only by about three percent.

Hydrazine monopropellant rocket motors have been built in a variety of sizes. The smallest is a vernier jet with a thrust of about 50 grams and a fuel consumption of 0.0005 lb/ms. At a depth of 20,000 feet it should generate about a tenth of a pound of buoyancy per minute. The largest available design burns 2.5 lb/ms to produce a quarter ton of thrust. At 20,000 feet it should generate buoyancy at a rate of almost 500 pounds per minute. Intermediate sizes are available to meet other requirements.

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*A given amount of hydrazine, with the water blend and reactor dimensions also given, will produce a fixed amount of gas, whether the system is open or closed. In a closed system, the water and undissociated ammonia will reduce the volume available for the gas. In other words, the presence of the water and ammonia inside the control volume degrades the specific density. In an open system, the water and undissociated ammonia pass through unconsumed, so that extra fuel tankage must be provided for them. Otherwise, they make no significant difference.

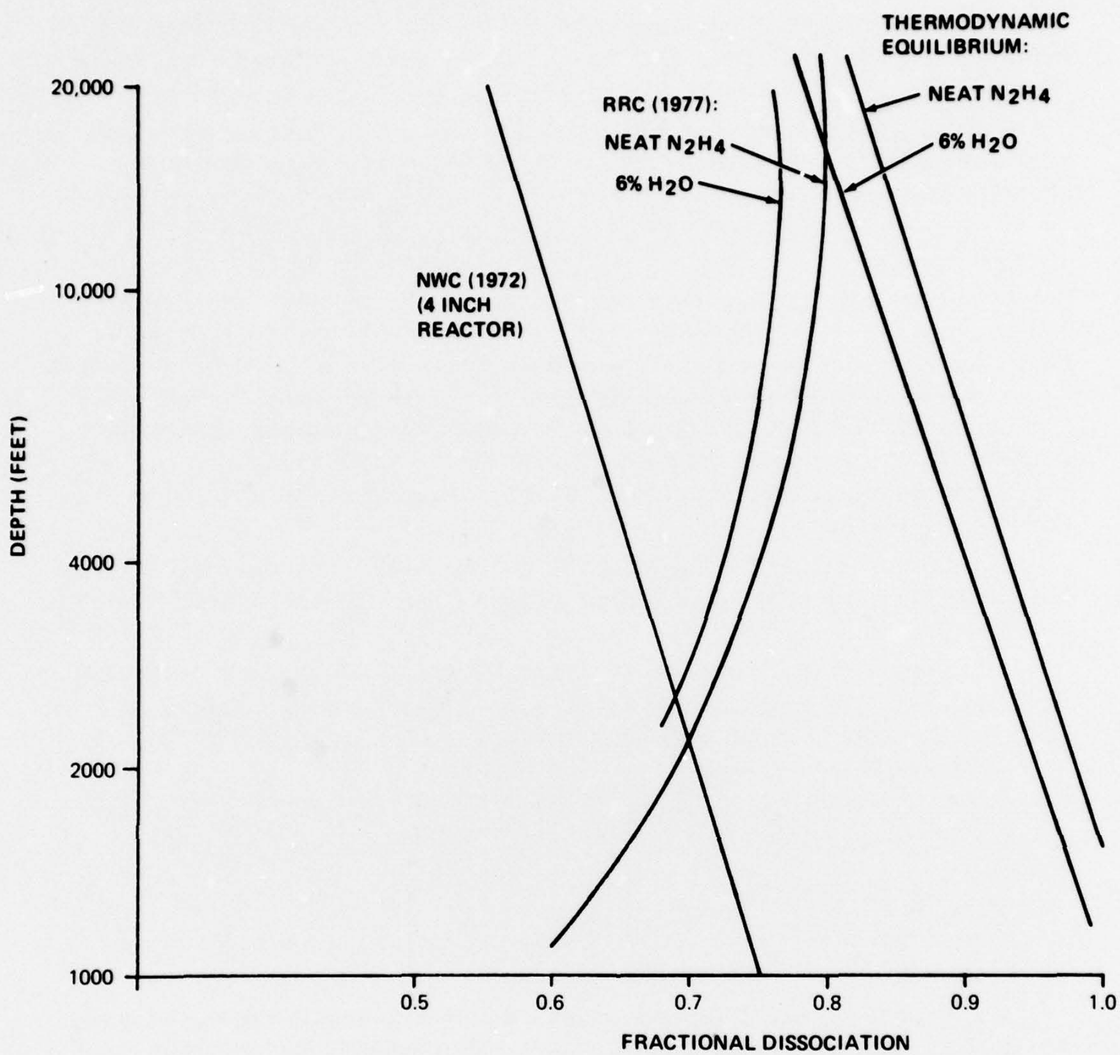

Figure 25. Dissociation efficiency for hydrazine.

DEVELOPABLE GAS SOURCES

Hydrogen is the ideal buoyancy gas, if a practical delivery method can be found. Even at 20,000 feet it weighs less than two and a half pounds per cubic foot. Two methods of hydrogen generation are under active investigation at this time.

Lithium Hydride

Certain metal hydrides have long been known as possible hydrogen generators. In 1952, Hurd (reference 15) identified the hydrides of lithium, sodium, calcium, lithium-boron, and sodium-boron as potentially useful in this respect. Recent studies have eliminated all but lithium hydride. The others are either inferior to hydrazine as a buoyancy generator, not well-behaved in seawater, or are expensive.

Lithium hydride has a specific gravity of 0.78 and a mole weight of 7.95 pounds. It reacts with water to produce 45.05 standard cubic feet of hydrogen per pound:

$$LiH + H_2O \rightarrow H_2 + LiOH + 47.560\ BTU$$

At 20,000 feet, 6.97 pounds of displacement per pound of lithium occurs. There is a small deduction for the loss of the buoyancy of the reacted lithium hydride. The weight of the hydrogen, 0.254 pounds, must also be deducted. The net buoyancy is 6.22 pounds per pound of lithium hydride. This may be compared with 3.17 pounds of buoyancy per pound of hydrazine in a six-percent water blend at 20,000 feet. Thus, lithium hydride is twice as efficient as hydrazine at 20,000 feet on a weight basis.

The reaction consumes nearly nine pounds of water to generate one pound of hydrogen. Excess water must be supplied if the reaction is to take place at a reasonable rate. This excess water must be on the order of six to 25 times the normal proportional amount. The excess water carries off the lithium hydroxide as fast as it is formed, and serves the additional function of carrying off the large amount of heat liberated. The temperature rise will be over 100°F, even with 25 times excess water.

The Naval Explosive Ordnance Disposal Facility (NAVEODFAC) developed an experimental system (reference 16), using five pounds of lithium hydride. Reaction time varied from eight to 50 minutes and was a strong function of the amount of excess water as well as of the geometric arrangement of the fuel. It is possible to build a hydrogen generator with a very short reaction time, but a design must be proved experimentally.

Lithium hydride is expensive. The price, for lots of over 500 pounds, is $11.70 per pound if screened to a maximum particle size of either one inch or 30 mesh; $15.68 per pound if "dust-free," screened to a minimum of 40 mesh and a maximum of three mesh. This may be compared with hydrazine at $3.40 per pound. Lithium hydride has several advantages, however. As already noted, it produces nearly twice as much buoyancy per pound as hydrazine at 20,000 feet. This buoyancy is in the most convenient form: a hydrogen-filled pontoon is the most compact form of buoyancy that can be constructed. As it rises, a hydrogen-filled pontoon will develop much less incremental buoyancy than if it were filled with any other gas. Finally, lithium hydride is safe to handle, compared to hydrazine. Only gloves and a dust mask are necessary, in addition to protecting the lithium hydride from moisture. Additional research is being conducted at this time on BRH, a compound of lithium hydride in a butyl rubber matrix. This compound is expected to have mechanical properties superior to the bulk material.

Supercorroding Alloy

The Civil Engineering Laboratory (CEL) has recently developed another possible source of hydrogen for buoyancy generation (reference 17). This is a so-called supercorroding alloy of magnesium and iron. The anodic and cathodic metals in powdered form are mechanically alloyed together in a high-energy ball mill. In seawater, the alloy corrodes away to produce hydrogen and heat. The theoretical yield is 14.72 cubic feet of hydrogen per pound of magnesium under standard conditions. The supercorroding alloy produces approximately 90 percent of the theoretical value for its magnesium content in a very short time. The reaction then tapers off. The fastest reaction rate is obtained with ten atomic percent of iron. The reaction is substantially complete in a very few minutes, depending on the temperature.

The alloy was originally developed as a self-contained heat source to relieve the extreme discomfort of divers in cold water. The heat of reaction is 84.36 kilocalories per gram-mole, or 156,074 BTU per pound-mole. This heat must be controlled if the

reaction is not to proceed with almost explosive speed. In the proposed diver heater, the reaction takes place in a chamber to which seawater is admitted in small, carefully-regulated amounts. It is assumed that a hydrogen generator would be quite different, with a small amount of the powdered alloy in a large container of seawater. The excess water must be 120 times the normal proportional amount if the temperature rise is not to exceed 36°F.

Pressure tests have yet to yield reproducible results, but there is some indication that the reaction rate is appreciably faster at pressures, such as those of the deep ocean, that prevent the formation of microscopic steam bubbles.

The supercorroding alloy has a third possible application. It can be compacted and sintered to form reliable self-destructing pins and links. The compaction and sintering processes, together with the formulation of the alloy and the diameter of the finished piece, permit control of both mechanical strength and release time. The devices currently used for anchor releases and such lack dependability.

The supercorroding alloy is entirely safe. It can be handled without danger, and can even come in contact with fresh water, since the reaction takes place only in the presence of an electrolyte such as seawater.

Lithium hydride is 3.4 times more efficient as a source of hydrogen, on a weight basis, than the alloy. Nevertheless, the alloy may merit development at least through the pilot plant stage. It could be relatively economical, even though the present cost is a nominal $100 per pound in one-pound lots. The raw materials are abundant and inexpensive, unlike lithium. And, unlike the process of manufacturing anhydrous hydrazine, the process could be efficient on a suitable scale.

SOLID MONOPROPELLANT

A final possible source of buoyancy gas is a solid monopropellant rocket. This concept has been carried through the laboratory stage. The Winchester Engineering and Development Group of Olin Corporation has conducted screening evaluations of various composite monopropellants at high pressure.

Most monopropellants are unsuitable for this application unless rapid generation of buoyancy is required. The burning rate starts off as an exponential function of chamber pressure:

$$r = aP_c^{\,n}$$

where P_c is the chamber pressure, and a is a constant. The reaction tends to bootstrap itself to the point where it is controlled only by the sonic flow of the exhaust gases. The exponent n is of particular interest. If it is greater than one, the reaction will almost certainly proceed in this manner, particularly if the initial pressure is equal to that at the ocean depth. Most solid monopropellants do have a pressure exponent greater than one.

One composite, a mixture of cellulose acetate and ammonium nitrate called LFT-6, passed the screening test. Its pressure exponent is still positive, but less than one. Formed into a hollow cylindrical grain five inches in length, with internal and external diameters of 0.5 inch and 2.8 inches, it performed well at pressures up to 10,000 psi.

The solid propellant is not an efficient source of buoyancy gas. The available data on the volume and composition of the gas, plotted in figure 18, indicate that its density is between nitrogen and decomposed hydrazine. On the other hand, the solid propellant has long shelf life, convenient operation, and manufacture from inexpensive materials.

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* TN's are informal documents intended chiefly for internal use.

APPENDIX A
STEPPED CABLE

Determine a combination of sizes of wire rope to lift 2,000 pounds from 20,000 feet. Maintain a safety factor of three based on ultimate tensile strength and the combined weight in water of object and cable.

Use steel-core hoisting rope. From table 1, Cw=1.61. From table 2, Cuw=Cva=9.0 × 10^4. The weights per foot and ultimate tensile strengths of various sizes are as follows:

Diameter	Pounds per foot	Strength, pounds
5/16	0.157	8,789
3/8	0.226	12,656
7/16	0.308	17,227
1/2	0.4025	22,500

For each segment, the allowable load at the upper end is the strength divided by the safety factor, and the load at the bottom end is known. The difference between the two loadings is the allowable weight of the segment, and this divided by the weight per foot gives the allowable length. Starting at the bottom:

$$\frac{\left(\frac{8789}{3} - 2000\right)}{0.157} = 5921 \text{ feet}$$

Round to 5,000 feet of 5/16-inch wire rope. This will weigh 785 pounds, and the load on the lower end of the next segment is 2785 pounds.

$$\frac{\left(\frac{12{,}656}{3} - 2785\right)}{0.226} = 6344 \text{ feet}$$

Round to 6,000 feet of 3/8-inch wire rope, weighing 1356 pounds. The total load is now 4141 pounds.

$$\frac{\left(\frac{17{,}227}{3} - 4141\right)}{0.308} = 5199 \text{ feet}$$

Round to 5,000 feet of 7/16-inch wire rope, weighing 1540 pounds. The total load is now 5681 pounds.

$$\frac{\left(\frac{22{,}500}{3} - 5681\right)}{0.4025} = 4519 \text{ feet}$$

Round to 4,500 feet of 1/2-inch wire rope, weighing 1811 pounds.

We have 20,500 feet of wire rope, allowing an extra 500 feet to provide some scope on the bottom. The cable weighs 5,492 pounds in water, 2.75 times as much as the object to be recovered.

APPENDIX B
FORCING FUNCTION

The sample vessel is a tug-supply boat with a length of 168 feet, a beam of 40 feet, and a draft of 13 feet. Two crane locations are considered. In the first, the crane is 98 feet aft, near the center of mass. In the second location, it is ten feet from the stern. In both cases, the crane is offset ten feet from the centerline to keep clear of the towing hawser. The boom is 25 feet long and ten feet above the deck. The centrally-located crane works over the side, 35 feet from the centerline, where it is affected by roll and heave. The stern-mounted crane works over the stern, and is affected principally by pitch.

Table B-1 is in terms of the significant wave height, defined as the average wave height, where the lower two-thirds of the waves are disregarded. Thus, the values in table B-1 are typical of the peak readings measurable at any given instant. A significant wave height of five feet is almost the upper limit of sea state three, and seven feet almost the upper limit of sea state four. Significant wave height for sea state five ranges from under eight to thirteen feet; ten feet represents an average sea state five.

The safety factor of the lift line depends on the highest value of boom tip motion encountered in the course of the lift. This highest value is expected to be a certain number

Table B-1. Significant vertical motions of boomtip for a 168-foot workboat.

Wave Height (*Significant*)	Displacement (*Ft*)	Velocity (*Ft/Sec*)	Acceleration (*g*)
1. Crane near centroid. Bows to sea.			
5.0	0.76	0.71	0.02
7.0	1.61	1.32	0.04
10.0	3.11	2.26	0.06
2. Crane near centroid. Beam sea.			
5.0	5.05	5.78	0.22
7.0	6.77	7.33	0.27
10.0	8.53	8.65	0.30
3. Crane on Stern. Bows to sea.			
5.0	3.35	3.33	0.11
7.0	5.44	5.03	0.15
10.0	8.02	6.85	0.19
4. Crane on stern. Beam sea.			
5.0	2.37	2.47	0.08
7.0	3.60	3.49	0.11
10.0	5.22	4.59	0.14
5. Crane on stern. Quartering sea.			
5.0	3.70	3.95	0.14
7.0	5.43	5.41	0.18
10.0	7.50	6.86	0.21

of times greater than the typical value in table B-1, depending on the duration of the operation. The multiple is given in figure 5 of the main text. Thus, for sea state five and a significant wave height of ten feet, the peak value in a three-hour period is expected to be two times the value in table B-1. For a duration of 24 hours the multiplier is 2.25, and for a week it is 2.45.

Table B-1 is for a particular vessel, but the values in it are fairly representative of the general category of vessels available for salvage and recovery operations. Vertical acceleration of the boom tip is typically in the range of 0.2 to 0.3 g for a variety of headings and weather conditions. When allowance is made for the duration of the lift, the peak values of vertical acceleration may be expected to lie in the general range of 0.6 to 0.7 g. In summary, then, even if the object to be lifted were two-blocked at the boom tip, the maximum dynamic load to be expected during the lift might lie in the range of 1.6 to 1.7 or more times the static load.

APPENDIX C
RESONANCE

Suppose that a 20,000-foot lift is being conducted in the open ocean using a Kevlar cable from a ship with a six-second roll period. If the safety factor is calculated as somewhere in the range of eight to ten, based on the assumed static load, the natural period of the object and line will also be about six seconds. The object and line will go into resonance and the safety factor will be nullified.

Referring to figures 8 and 10 of the main text, reducing the safety factor would merely result in resonance at a shallower depth. The line probably would fail before the object reached the surface. From figure 10 it may be seen that the exact length for resonance with a safety factor of three is 7,500 feet. In this case, the weight-in-water of the line would be reduced as the object was raised and the safety factor would increase slightly. The exact length for resonance would be 7,750 feet, where the safety factor would be 3.074. Failure would probably occur before this point was reached.

APPENDIX D
DYNAMIC MASS

In the recovery of an aircraft, the added mass may be the largest component of the dynamic mass. Consider an aircraft fuselage suspended horizontally in water. In figure 12 of the text, adapted from Holmes (reference 2) and Lamb (reference 3), there is a curve for the added mass of a cylinder being accelerated perpendicular to its axis. The abscissa in this case is the ratio of the length of the cylinder to the diameter. The cylinder can be considered to represent a fuselage. If the length of the fuselage is six times the diameter, figure 12 shows that the added mass is a volume of water equal to 0.925 times the volume of the fuselage.

The added mass is even more significant for a wing suspended horizontally. In the figure, there is also a curve for thin plates being accelerated broadside. A wing may be considered just such a thin plate. If the aspect ratio of the wing is 8.5, the added mass is equal to the mass of water in a cylinder of a diameter equal to the wing chord and a length equal to the wing span.

Refer to table D-1. The effective mass is calculated for a C–130 transport, assumed to be resting intact on the ocean floor. The calculations are approximate, based on dimensions and drawings published in Jane's All the World's Aircraft. As the wing can carry 930 cubic feet of fuel, it is assumed to have a floodable volume of 1,800 cubic feet for the two wings. With an assumed cargo weight of 25,000 pounds, and an assumed overall density equal to that of aluminum (0.10 pound per cubic inch), the aircraft is estimated to weigh 63,171 pounds in seawater. If raised in a level attitude, the effective mass is almost thirteen hundred tons, or forty-five times the weight in seawater. Even if raised nose first or tail first, the effective mass is fourteen times the weight in seawater.

Figures 9, 10 and 11 of the text are easily corrected for this effective mass. Let L_o and T_o be the resonant length and period from figure 9, 10 or 11, while L and T represent the true resonant length and period to be determined. Let the weight in water be M_o and the effective mass be M. Enter figure D–1 with M/M_o to read L/L_o against T/T_o.

Suppose that the C–130 is to be raised in a level attitude from 10,000 feet, employing a nylon 2-in-1 line. A nylon 2-in-1 line of four-inch diameter will have an estimated static safety factor of 6.06. Suppose further that twelve seconds is the longest wave period of concern for the operating area, the season, and the equipment to be deployed. From figure 11, L_o is 1,650 feet if T_o is 12 seconds. From figure D–1, choosing T equals T_o and M/M_o is 50, L/L_o is 0.0225. Thus, L is 37 feet. The aircraft can be raised safely with the nylon line to a depth of about 65 feet, at which point SCUBA divers can attach flotation devices to bring it to the surface.

Table D-1. Dynamic mass of C-130 transport.

1. Dimensions

Wing	Fuselage	Tailplane
Span 132.5 ft	Length 97.75 ft	Span 52.7 ft
Root chord 16.0 ft	Max Diameter 13.66 ft	Area 381 ft^2
Mean chord 13.75 ft	Equiv. Length 71 ft	Mean chord 7.23 ft
Area 1745 ft^2	L/D 5.2	L/W 7.29
Equivalent L/W 9.23		

2. Weight in Air

Wing	Fuselage	Tailplane
Operating weight empty		75,331 lb.
Assumed weight of cargo		25,000 lb.
Total		100,331 lb.

3. Weight in Seawater

Wing	Fuselage	Tailplane
Assume all fuel leaked out, and an overall specific gravity equal to that for aluminum (2.7 with respect to seawater)		
Assumed weight in seawater		63,171 lb.

4. Contained Mass

	Volume	Mass
Wing	1,800	115,200 lbs.
Fuselage	10,400	665,600 lbs.
Tailplane	negl.	negl.
Total	12,200	779,800 lbs.

5. Added Mass

	Coefficient	Volume	Mass
Wing	1.00	18,843 ft.	1,205,970 lbs.
Fuselage	0.91	9,464 ft.	605,696 lbs.
Tailplane	0.99	2,142 ft.	137,086 lbs.
Total		30,449 ft.	1,948,752 lbs.

6. Totals

	If raised in a nose-first or tail-first attitude, the added mass may be neglected.	If raised in a level attitude,
Weight in air	100,331 lbs.	880,131 lbs.
Contained Mass	779,800 lbs.	1,948,752 lbs.
Effective Mass	880,131 lbs.	2,828,883 lbs.
	13.9 X wt. in seawater	44.8 X wt. in seawater

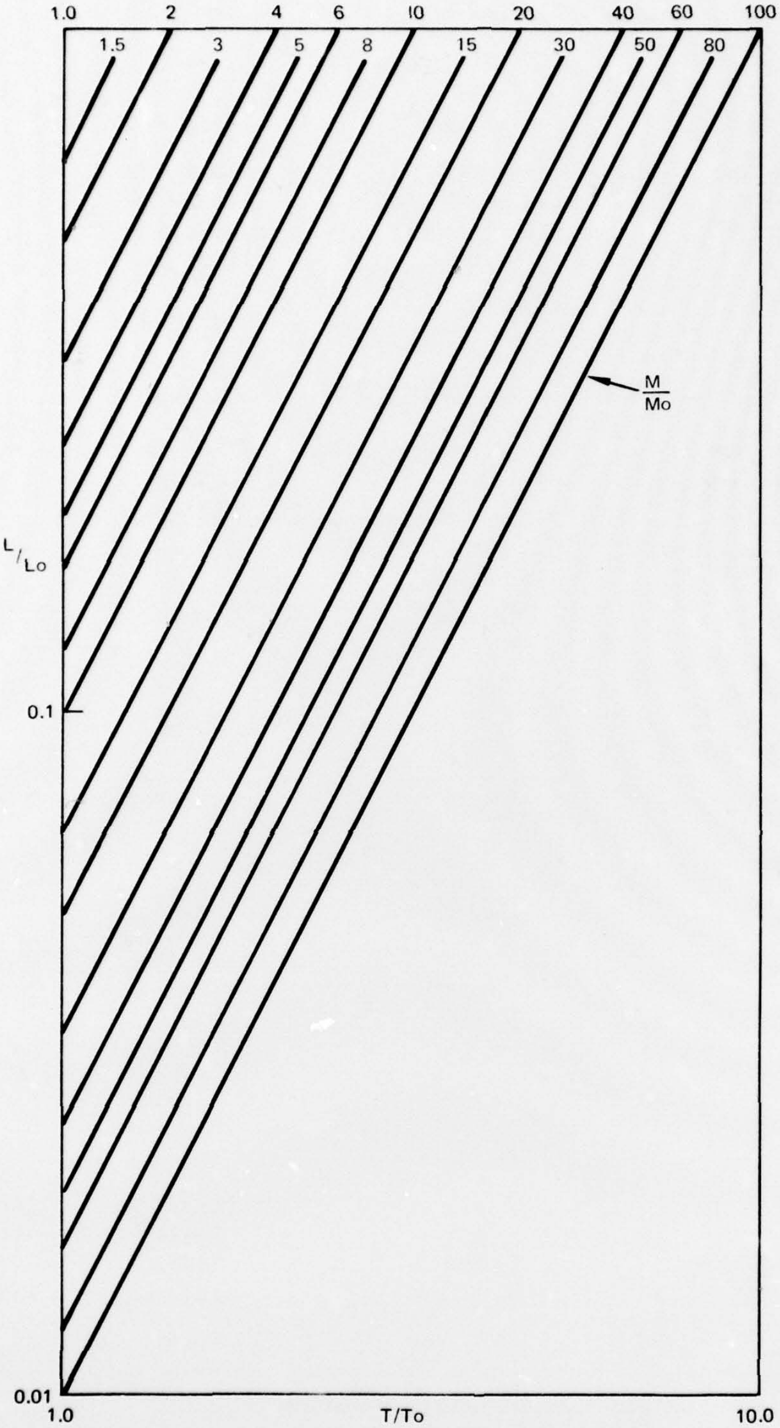

Figure D-1. Period, mass and length ratios for resonance.

APPENDIX E
CALCULATING THE LIFT CAPABILITY OF DRILL SHIPS

Consider a drill string made from V150-grade steel with a minimum yield strength of 150,000 psi. Drill string 9 5/8 inches in diameter and 0.545 inches thick has an allowable tensile load of 1,166,000 pounds, or 583 tons, with a safety factor of two. This is within the capabilities of such ships. The string weighs 53.5 pounds per foot in air, including couplings, and 46.5 pounds per foot in seawater. A string 20,000 feet long weighs 465 tons, leaving 118 tons as the residual allowable load on the string. Thus, the net lifting capability will be 118 tons, provided the ship can accommodate 583 tons.

In this example, the net lift capability at 20,000 feet is 20 percent of the gross lift capability. The net lift capability decreases linearly with depth at the rate of four percent per thousand feet.

This performance can be improved in two ways, both of which were demonstrated on GLOMAR EXPLORER. The diameter of the pipe can be stepped, and a stronger grade of steel can be employed. GLOMAR EXPLORER has six sizes of pipe to reach 17,000 feet. The pipe is gun barrel steel, with an allowable tensile stress of 106,000 psi after a safety factor of 1.5 has been applied to the ultimate tensile stress. The net lift capability decreases with a logarithmic decrement of four percent per thousand feet. If outfitted for 20,000 feet, GLOMAR EXPLORER's net lift capability at this depth would be 3,840 tons including the weight of the attachment mechanism, or 45 percent of the gross capability.

APPENDIX F
SAMPLE CALCULATION FOR SPHERICAL PRESSURE BALLAST TANK

Assume sphere is to lift one long ton from 20,000 feet. Sphere is titanium with a tensile strength σ = 150,000 psi. Design safety factor is 2 based on theory for near-perfect sphere.

Allow 2% extra buoyancy for the weight of the complete system.

Buoyancy B = 1.2 X 2240 = 2688 lbs. = 42 ft.^{3}

Pressure at working depth P = 8888 psig

Sphere will be ballasted at surface with 2240 lbs or 35 ft^3 of seawater, then pressurized with hydrogen gas to some internal pressure p_o. Internal pressure will drop during deployment, first from cooling and then from expansion as ballast is pumped out.

Assume that the final internal pressure is equal to 48% of p_o. Then, if the positive pressure differential on deck is equal to the negative pressure differential when deballasted at depth,

$$p_o = P - 0.48 p_o$$

$$p_o = 0.675\ P = 6{,}000 \text{ psig.}$$

Multiplying by the safety factor of 2, the collapse pressure is

$$P_D = 12{,}000 \text{ psig}$$

which is equivalent to designing an unpressurized sphere for a collapse depth of 27,000 feet.

Failure will be by yielding. The thickness ratio

$$k = \frac{P_D}{2\sigma} = \frac{12{,}000}{3 \times 10^5} = 0.04$$

$$\rho_s = \frac{276}{64}(3k - 3k^2 + k^3) = 0.497$$

which checks with figure D-1.

Displacement $D = \frac{B}{1-\rho_s} = 5344 \text{ lb} = 83.5 \text{ ft}^3$

Weight in air $= \rho_s D = 2656$ lb

Outside diameter d = 5.42 ft. = 65.07 in.

Thickness $= k\,\frac{d}{2} = 1.30$ in.

Internal diameter $d_i = (1-k)d = 62.47$ in. =5.21 ft.

Internal volume $V_i = \frac{\pi}{6} d_i^3 = 73.86 \text{ ft}^3$

After ballasting, the residual volume

$$V_1 = 38.86 \text{ ft}^3.$$

After ballasting and prior to deployment, the sphere is charged to 6000 psig, with hydrogen gas at ambient temperature, assumed to be 70°F. The weight of the gas is about 55 lb.

When the sphere is deployed to 20,000 feet, it cools to the new ambient temperature, assumed to be 29°F, and the internal pressure drops to

$$P_1 = \frac{489}{530} \times 6000 = 5536 \text{ psig.}$$

As the sphere is deballasted, the internal pressure drops to a final value of

$$\frac{38.86}{73.86} \times 5536 = 2913 \text{ psig.}$$

The differential pressure is now 5975 psig, within the margin of safety

During the deballasting, the internal pressure

$$P_i = \frac{V_1}{V_1 + V} P_1$$

where V is the volume of ballast water already pumped out. The energy of pumping is

$$E = \int_o^V \left[P - \frac{V_1}{V_1 + v} P_1 \right] dv$$

$$= PV - P_i V_1 \ln\left(\frac{V_1 + V}{V_1}\right)$$

$$= 8888 \frac{\text{lbF}}{\text{in}^2} \times 144 \frac{\text{in}^2}{\text{ft}^2} \times 35 \text{ ft}^3$$

$$- 5536 \frac{\text{lbF}}{\text{in}^2} \times 144 \frac{\text{in}^2}{\text{ft}^2} \times 38.86 \text{ ft}^3 \ln\left(\frac{73.86}{38.86}\right)$$

= 44,795,520 ft. lb – 19,894,623 ft. lb.

= 24,900,981 ft. lb.

= 12.58 hp-hr

= 9.38 kWh

Assuming on overall efficiency of 65% for the electric motor and deballasting pump, 14.43 kWh of electrical energy is required to deballast the sphere, or 290 pounds of silver-zinc batteries.

The allowance for extra system weight was 448 lb. We have

hydrogen gas	55 lb.
batteries	290 lb.
motor pump, misc.	103 lb.